

AD-A143 971

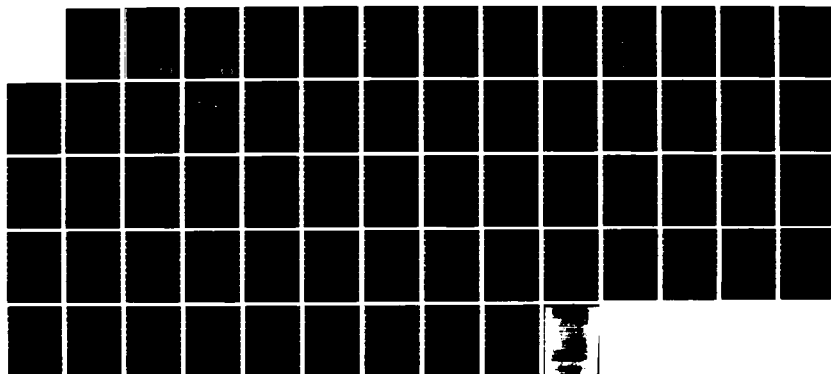
EVALUATION OF SERVICE LUBRICANTS AND METALS CORRELATION 1/1  
OF BENCH TESTS WIL. (U). IMPERIAL COLL OF SCIENCE AND  
TECHNOLOGY LONDON (ENGLAND) LUBR. T NONAKA ET AL.

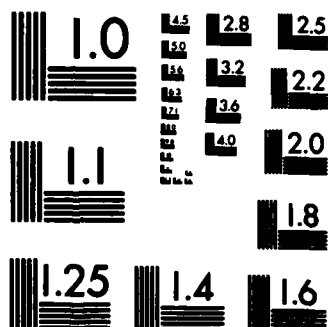
UNCLASSIFIED

JUN 84 DAJA45-83-C-0001

F/G 11/6

NL





MICROCOPY RESOLUTION TEST CHART  
NATIONAL BUREAU OF STANDARDS-1963-A

AD-A143 971

12

AD

EVALUATION OF SERVICE LUBRICANTS AND METALS  
CORRELATION OF BENCH TESTS WITH ENGINES

Final Technical Report

by

T. Nonaka and A. Cameron

June 1984

United States Army  
EUROPEAN RESEARCH OFFICE OF THE US ARMY  
London England

CONTRACT NUMBER DAJA 45 83 C 0001

Lubrication Laboratory,  
Department of Mechanical Engineering  
Imperial College of Science & Technology, London, SW7

Approved for Public Release: distribution unlimited

DTIC FILE COPY

DTIC  
ELECTE

AUG 1 1984

B

AD

EVALUATION OF SERVICE LUBRICANTS AND METALS  
CORRELATION OF BENCH TESTS WITH ENGINES

Final Technical Report

by

T.Nonaka and A.Cameron

June 1984

United States Army  
EUROPEAN RESEARCH OFFICE OF THE US ARMY  
London England

CONTRACT NUMBER DAJA 45 83 C 0001

Lubrication Laboratory,  
Department of Mechanical Engineering  
Imperial College of Science & Technology, London, SW7

Approved for Public Release: distribution unlimited

DTIC  
ELECTE  
AUG 1 1984  
S B

UNCLASSIFIED

R&amp;D 3049-AN

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO. AD-A143971	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Evaluation of Service Lubricants and Metals CORRELATION OF BENCH TESTS WITH ENGINES		5. TYPE OF REPORT & PERIOD COVERED Final Report
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) T. Nonaka & A. Cameron		8. CONTRACT OR GRANT NUMBER(s) DAJA45-83-C-0001
9. PERFORMING ORGANIZATION NAME AND ADDRESS Lubrication Laboratory, Imperial College London		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 6.11.02A IT161102BH57-06
11. CONTROLLING OFFICE NAME AND ADDRESS  USARDSG-UK Box 65, FPO NY 09510		12. REPORT DATE June 1984
		13. NUMBER OF PAGES 21
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report)  Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE
16. DISTRIBUTION STATEMENT (of this Report)  Approved for public release; distribution unlimited		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number)  Diesel Engine. Liner Scuffing. Bore Polish. Bench Test  Anti wear. Extreme Pressure. HFR Test. Correlation tests:		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number)  The object of the work, to devise a test that enables engine lubrication to be correlated by the High Frequency Reciprocating (HFR) test apparatus, recently developed in the Lubrication Laboratory of Imperial College, has been successfully completed. Four oils whose performance were evaluated in the full scale 240 hour diesel scuffing test, 6V 53T, have been shown to fit in with the HFR test. A further eleven oils which had been tested in the Ford Tornado bore polish sequence were also (Cont'd)		

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

20. Cont'd

satisfactorily rated. Of these eleven, 5 had also been run in the Volvo TD 120A test. The Volvo rated them differently from the Ford Tornado, which the HFR confirmed.

Calcium detergent was found to have a considerable effect on wear.

Most of the additives tested gave a surface film characterized by a high electrical contact resistance, ECR, which may reach  $10^6$  ohms. When it is present, there is very little or no wear. It is broken down mainly by temperature but also by load, with consequent increase of wear.

The project was primarily concerned with the influence of lubricants on engines, so standard engine liners were used. Some test, however, clearly demonstrated the importance of liner metallurgy and liner finish. This subject would repay further study. The importance of combustion atmosphere is briefly discussed.

This work shows that the state of the electrical contact resistance (ECR) layer is the key to satisfactory engine lubrication. It is desirable to continue to study its properties in more detail.

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

(i)

### SUMMARY

The object of the work, to devise a test that enables engine lubrication to be correlated by the High Frequency Reciprocating (HFR) test apparatus, recently developed in the Lubrication Laboratory of Imperial College, has been successfully completed. Four oils whose performance were evaluated in the full scale 240 hour diesel scuffing test, 6V 53T, have been shown to fit in with the HFR test. A further eleven oils which had been tested in the Ford Tornado bore polish sequence were also satisfactorily rated. Of these eleven, 7 had also been run in the Volvo TD 120A test. The Volvo rated them differently from the Ford Tornado, which the HFR confirmed.

Calcium detergent was found to have a considerable effect on wear.

Most of the oils investigated gave a surface film characterised by a high electrical contact resistance, ECR, which may reach  $10^6$  ohms. When it is present, there is very little or no wear. It is broken down mainly by temperature but also by load, with consequent increase of wear.

The project was primarily concerned with the influence of lubricants on engines, so standard engine liners were used. Some tests, however, clearly demonstrated the importance of liner metallurgy and liner finish. This subject would repay further study. The importance of combustion atmosphere is briefly discussed.

This work shows that the state of the electrical contact resistance (ECR) layer is the key to satisfactory engine lubrication. It is desirable to continue to study its properties in more detail.

Accession For	
NTIS GRA&I	<input checked="checked" type="checkbox"/>
DTIC TAB	<input type="checkbox"/>
Unannounced	<input type="checkbox"/>
Justification	
PER CALL TO	
By	
Distribution/	
Availability Codes	
Dist	Avail and/or Special
A-1	



## CONTENTS

	page
Summary	(i)
Object	1
Form of Report	1
PART 1. TEST APPARATUS AND PROCEDURES	2
(i) Design Concept	2
(ii) Description of Rig	3
(iii) Vibration	4
(iv) Test Specimens	4
(v) Test Lubricants	5
(vi) Test Procedure	5
(vii) Scuffing Test	6
Measurement of Wear	6
PART 2. CORRELATION WITH ENGINE TESTS	7
Introduction	7
Wear Sequence 1	7
Wear Sequence 2 Drip Feed	7
Wear Sequence 3	9
Importance of Liner Material	9
6V 53T Scuffing Test	10
a) Wear	10
b) High Temperature	10
c) Effect of Load on Scuffing	11
PART 3. PROPERTIES OF HIGH ELECTRICAL CONTACT RESISTANCE FILM	12
Introduction	12
Lubricants	12
a) Additive Response	12
b) Base Stocks	13
Effect of Calcium Detergent	14
Effect of Load	14
a) Anti-wear, High ECR Region	14
b) Extreme Pressure REGION	15
PART 4. LINER METALLURGY AND FINISH. COMBUSTION ATMOSPHERE	16
Introduction	16
Silicon Carbide Impregnation	16
Used Oil and Liner	16
Combustion Atmosphere	17
Conclusions	18
Acknowledgements	18
References	19
List of Figures	20
Appendices	
Details of Oils	
Details of Liners and Rings	



## Object

The prime object of this work is quite simple. A test device, the High Frequency Reciprocating (HFR) machine, had been developed in the Lubrication Laboratory at Imperial College, London, which showed promise in giving good correlation with engine performance. In order to check and confirm this, oils whose performance had been rated in actual engine tests were obtained and run in the test machine. An unambiguous quantitative method of evaluation of the results from the device was part of the requirement.

The US Army Fuels and Lubricants Laboratory at San Antonio, Texas, supplied 6 oils, 4 of which had been run in the 240 hour 6V 53T scuffing test together with samples of liner and ring. Mobil Oil Co. Ltd. of London gave 6 fully formulated oils which had been run in the Ford Tornado 200 hour Bore Polishing sequence. Castrol furnished 5 oils which had been rated in both the Ford Tornado and the Volvo bore polishing test. The base oils of these last 5 were also available for test.

A secondary but most important object was to study the mechanism of scuffing and bore polishing. To this end a number of subsidiary tests were carried out.

The US Army was interested in rating the scuffing of the oils sent to the Laboratory. The correlation of the scuffing performance of these oils by the HFR was very rapidly and successfully achieved. The related problem, bore polishing, was then studied in detail.

## Form of the Report

Descriptions of the High Frequency Reciprocating (HFR) machine have been published a number of times. A brief resumé is given of the philosophy behind the design of the machine which was used here. A discussion of the methods used to overcome the problem of parasitic vibrations, which caused some trouble, are then described. This is followed by details of the test specimens and lubricants and then a considerable section on the development of the test methods. This forms the first part of the report.

It was decided to concentrate first on a quantitative wear test and correlate the results with bore polishing, as the test oils were all obtained from firms close to London (Mobil Ltd and Castrol). When these were satisfactory, greater attention was paid to the 4 US Army oils, for which 6V 53T scuffing data was available, and scuffing tests were carried out. This forms part two of this report. The third part describes various experiments designed to elucidate the type of lubrication found under these conditions.

Part 4 reports some tests on liner materials and finish and discusses the atmosphere of the test. In the appendices are details of the oils used and of the liners.

## PART 1. TEST APPARATUS AND PROCEDURES

### (1) Design Concept

The basic concept on which this test machine is based is as follows.

It is well known that the material from which cylinder liners and piston rings are made as well as their method of finish, honing, lapping, grinding, etc., are crucial in the successful operation of diesel engines. The type of lubricant and additive are equally important. This sensitivity is interpreted as showing that it is asperity contact which largely determines the successful running of these engines.

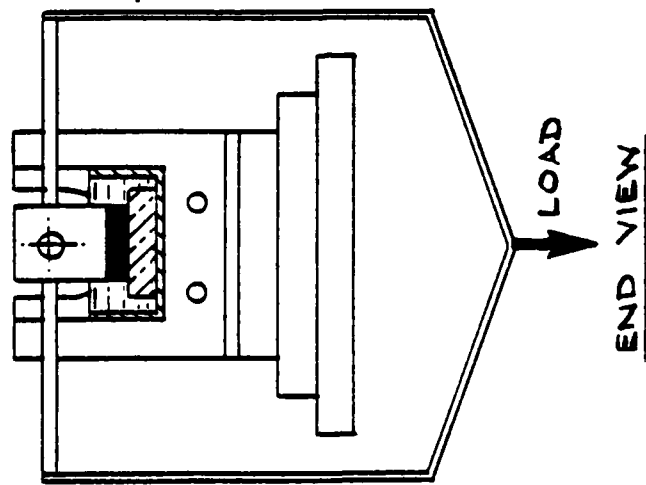
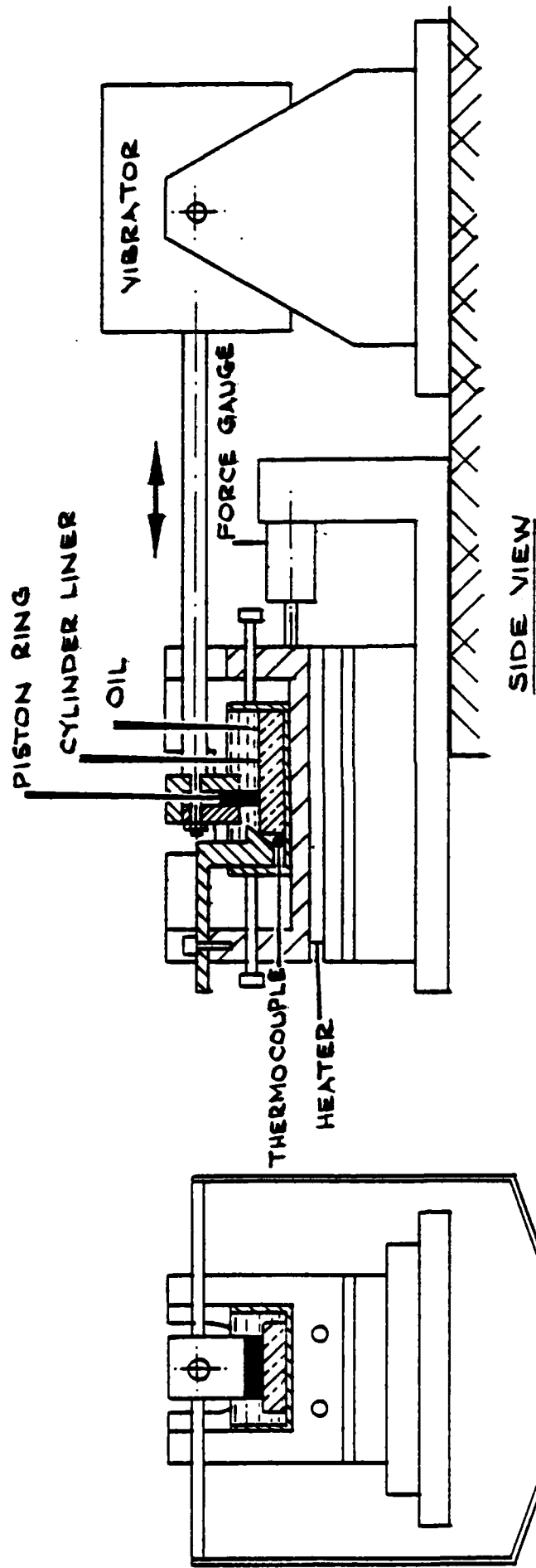
In order to investigate in the laboratory diesel engine lubrication, an apparatus has to be designed which reproduces accurately the type of boundary lubrication under which the asperities operate. Boundary lubrication is specified as this implies that it is the surface properties of the system which are decisive. Hydrodynamic lubrication on the other hand is solely determined by external bulk parameters, load, speed, viscosity. If hydrodynamic lubrication was the deciding factor, any type of fluid of a suitable viscosity, e.g. glycol or glycerine, would be as satisfactory as mineral oil, which quite obviously it is not.

The apparatus therefore is designed to exclude all normal hydrodynamics and reproduce as accurately as possible asperity lubrication. The following features were considered important.

1. Temperature at the contact must be the same in engine and test rig.
2. Since surface finish and material is so important, the test machine must use actual engine components. The environment must also be considered.
3. When two materials rub together, the oxide coating is removed, in whole or part. The test pieces must be loaded and move the one over the other, in frictional contact.
4. The active components in the lubricant react with the surfaces, and the rate of reaction is important. The repetition rate in test and engine must be matched.

These four requirements have been largely met in the design of the HFR machine.

In this device a small segment of ring is loaded against a piece of cylinder liner and reciprocated with a short stroke length at a rate similar to engine speed. Here 50 Hz is the standard speed. The specimens are heated to the required operating temperature. The load and stroke length are small enough to ensure that frictional heating is acceptably very small. Calculations show it is only about



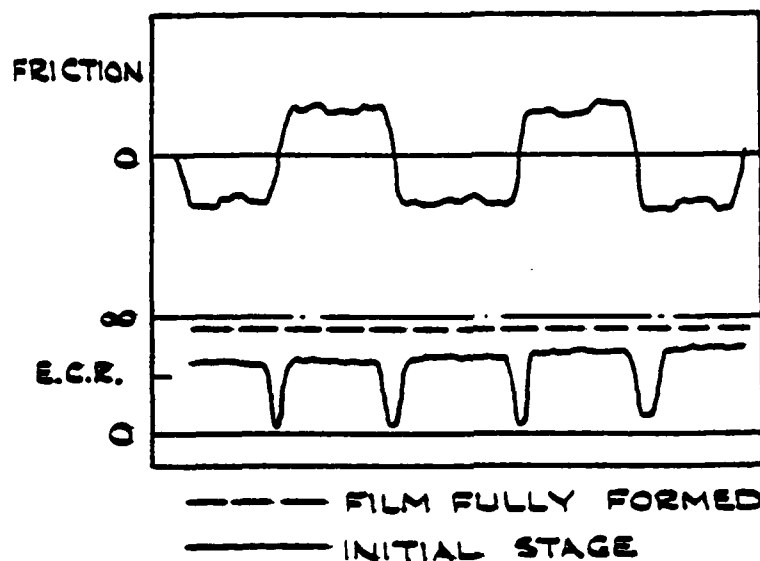
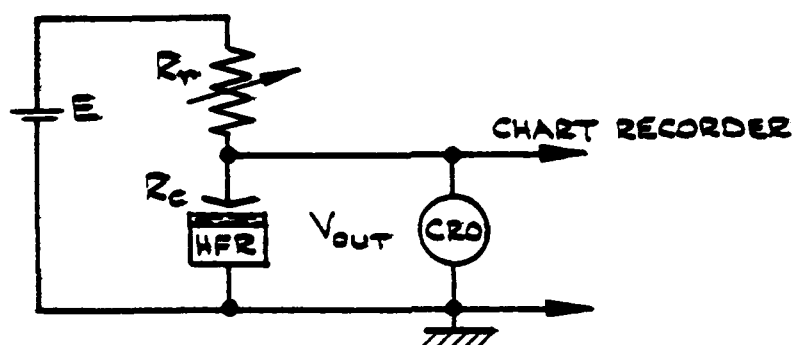


Fig. 2. Instantaneous Trace of Friction



E CONSTANT VOLTAGE SUPPLY 16mV

$R_r$  REFERENCE RESISTANCE 10 OHMS TO 100K.OHMS

$R_c$  EFFECTIVE CONTACT RESISTANCE

$V_{out}$  POTENTIAL DIFFERENCE ACROSS CONTACT  
AS MEASURED BY HIGH IMPEDANCE 'SCOPE

$$V_{out} = E \cdot \frac{R_c}{R_r + R_c} \quad \text{HENCE } R_c$$

Fig. 3. Circuit to measure Electrical Contact Resistance. ECR

5°C maximum.

A basic assumption in all this work is that whatever the total load the local stress on each asperity is the same, and equal to the yield stress of the material. For this reason a small load causes the same stress on each asperity as does the complete ring in the engine.

Another feature is that though the sliding velocities are not the same in the test rig and the engine the repetition rates are similar, and this is decisive.

One major difference, which will be discussed later, is that the test machine operates in the laboratory, not in combustion atmosphere.

Having discussed the concept, the actual design will now be described.

#### (ii) Description of Rig

The apparatus has been described a number of times. (Refs 1, 2 & 3) One member of the frictional pair, the dynamic specimen, is held by a clamp at one end of a rod whose other end was attached to a vibrator. This is a 25 watt electrical "loud speaker" type device driven by an oscillator and amplifier. (Figure 1)

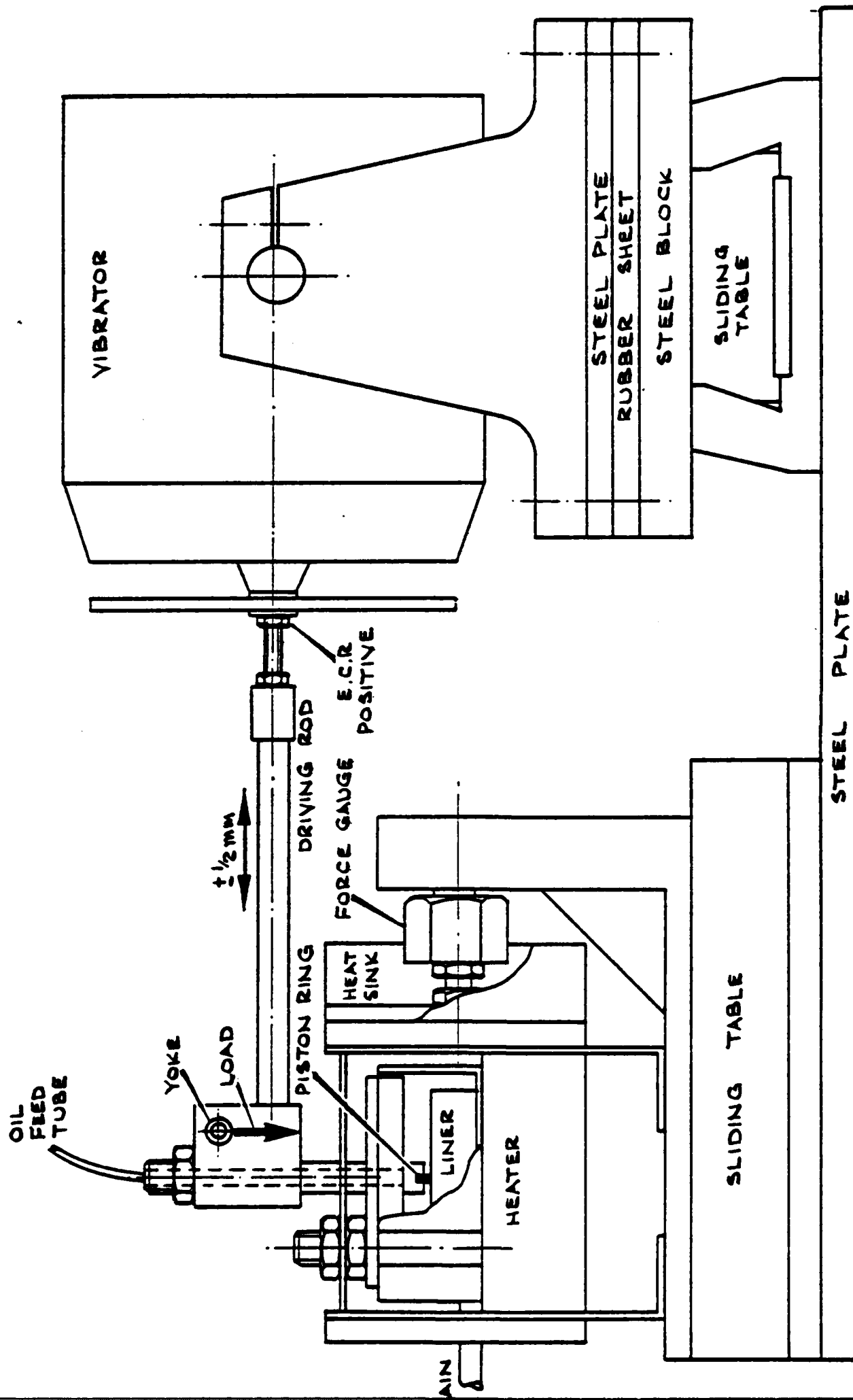
A stroke of length  $\pm 1\text{mm}$  was standard throughout the test, to avoid serious friction heating. However, it could be varied continuously up to  $\pm 1.5\text{mm}$ . The frequency could be any value up to 200Hz, a value of 50Hz was used. The dynamic specimen was loaded against the static specimen by a weight suspended directly below the point of contact.

The oil bath, after careful cleaning, was held tightly against a heater block, which held two 200 watt electrical heating rods. The heater block itself was mounted on a resilient mounting. Alternative mountings, e.g. Teflon pads and steel rollers have also been tried in an attempt to reduce resonance vibration.

A piezo-electric force gauge with charge amplifier, was used to measure the friction. Resonant forces naturally were seen in the output from the force gauge; these were reduced to a minimum. The high stiffness of a piezo electric crystal was needed to ensure the resonant frequency of the system was high. The output of the amplifier (monitored by a cathode ray tube) was rectified and fed to a chart recorder. The instantaneous friction trace from the CRO is shown in Figure 2.

The temperature of the specimen, measured by a thermocouple, placed within 5 mm of the contact, was also recorded.

Two methods of lubricant supply were used. In one the friction pair was immersed in the heated bath. It was found that the oil coked and changed its properties over a long test. In the other, oil was



dripped on to the contact through the tube holding the specimen. A perfusor, maximum rate 3.3 ml/hour, was used to supply the oil.

It was also considered that drip feed was much closer to the type of splash lubrication, which occurs on cylinder walls, than a totally immersed specimen.

By putting 16 mV across the contact and also across the input of a cathode ray oscilloscope, the conductivity of the film between the friction pair could be monitored. The circuit producing this potential is shown in Figure 3. When the 'scope shows half the applied potential the effective film resistance equals the source impedance.

The rate of temperature increase and constant set temperature were controlled by a micro processor. A major difficulty, which had been experienced by all previous test devices, was parasitic vibrations. The means used to overcome these are described next.

#### (iii) Vibration

Vibration caused a major problem. There were two sources for this.

- 1) the resonant frequency of the whole system
- 2) a sinusoidal wave, sometimes 30% of the signal, caused by interaction of the mass of the machine and the mass of the vibrator core. This spurious signal was cured by increasing the mass of the vibrator system

To achieve separation of vibrator and heater/specimen holder, each was placed on heavy blocks. These were on a thin (10mm) steel plate, which transmitted only very little vibration between them. The machine could be slid to achieve alignment. The vibrator was attached to its block by four screws. Mounting it on rubber with no fixing caused vibration.

The driving rod was aluminium, as short and light as possible. It gave less vibration than a steel rod used previously.

Coupling the force gauge directly to the heater block through the heat sink also reduced vibration. Rigidity of the system was the key to reducing vibration as was good alignment.

Supporting the heater block on rollers or Teflon (P.T.F.E.) sliding pads increased vibration. The modified design of the test machine is shown in Figure 4.

#### (iv) Test Specimens

In total, test specimens were cut from two piston rings A and B and cylinder liners from 7 engines. These were Ford Tornado

Volvo TD 120A, US Army test engine, 6V 53T, Komatsu and two from Rolls Royce.

The rings were chrome electro plated scraper rings. Their diameter was slightly smaller than the liners. Details are in the appendix.

No difference in performance was found between rings A and B if the wear on ring B was even.

(v) Test Lubricants

The main object of this work was to correlate the results of the HFR with actual engine performance. Three sets of oils which had been rated in engines were available to us. These were:-

- 1) Six oils from the US Army. Four had been rated for scuffing in the 6V 53T test.
- 2) Six lubricants tested in the 200 hour Ford Tornado bore polishing sequence (Mobil).
- 3) Five which had both Ford Tornado and Volvo polishing ratings. Furthermore, 31, the base stock of these 5, was also available (Castrol).
- 4) The solvent neutral base stock, 350 NS, from Elf Aquitaine used in previous work (Ref. 1).

Details are in the appendix.

(vi) Test Procedure

a) Preparing Specimens

The static test piece was a segment, 30mm x 15mm, taken from the liner. This was cut dry to avoid contamination by cutting oil, which is often sulphurised. This was an important precaution.

The moving specimen was a 10mm length of scraper ring.

b) Cleaning

All test pieces and any parts in contact with the test lubricant were cleaned ultra sonically first, for 10 minutes, in Toluene, then again for 10 minutes in Acetone, and finally air dried. They were assembled with tongs to prevent contamination.

c) Test Sequence

Wear tests were developed in three stages. These are as follows, labelled Wear 1, Wear 2 and Wear 3.



Wear 1

Load: 2lbf from start of test

Temperature: Heat from room temperature to 250°C at 7°C /min  
(30 mins). Held constant for a further 90 minutes.

Duration: 2 hours total

Lubrication: Immersed.

Wear 2

Load: 2½lbf constant

Temperature: Run-in for 5 minutes at room temperature.  
Heat at selected rate to test temperature  
Held constant for remainder of test, usually 7 hours.

Lubricant: Drip.

Wear 3

As for Wear 2 but no running-in and heat to test temperature with specimens out of contact.

(vii) Scuffing Test

Load: 3 lbf standard (½lbf to 6lbf also tested)

Temperature: Run in at room temperature 5 minutes. Heat to 350°C at 3°C/minute.

Duration: 2 hours approx.

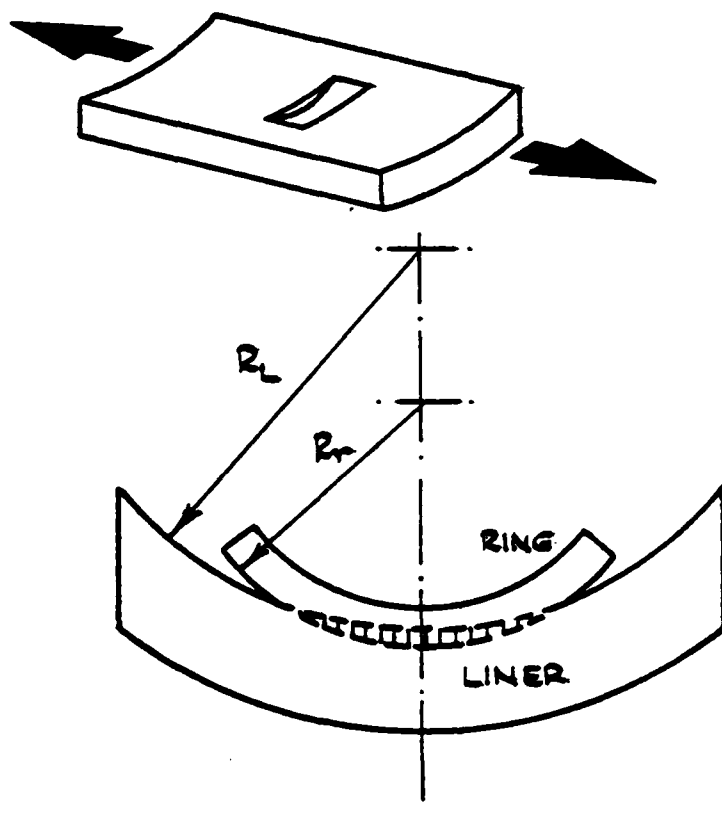
Lubrication: Drip.

In all 4 test procedures the frequency was 50Hz and stroke  $\pm \frac{1}{2}$ mm.  
The drip feed rate was 3.3 ml/hour.

The low heating rate of 3°C/min was chosen for the scuffing as it was found that friction takes a finite time to adjust to a new temperature.

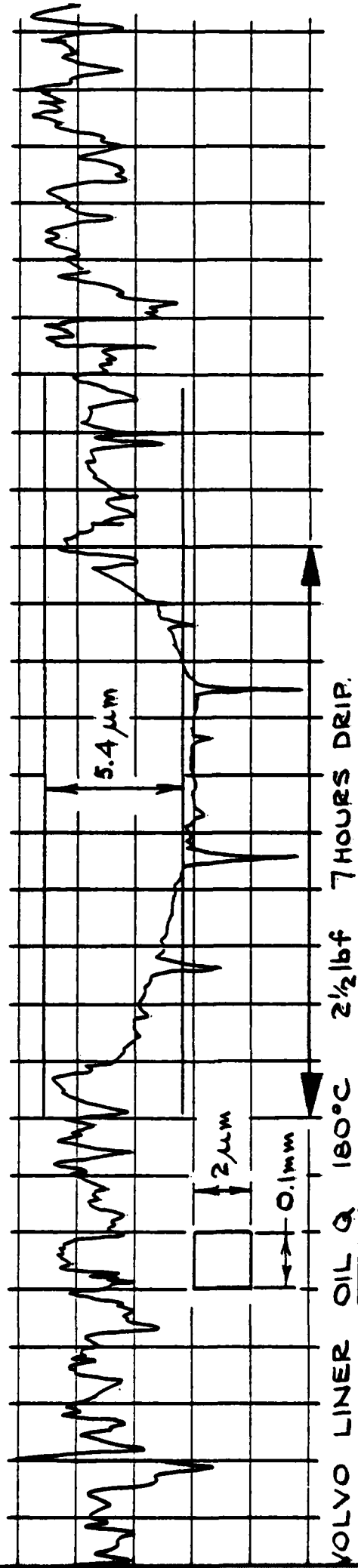
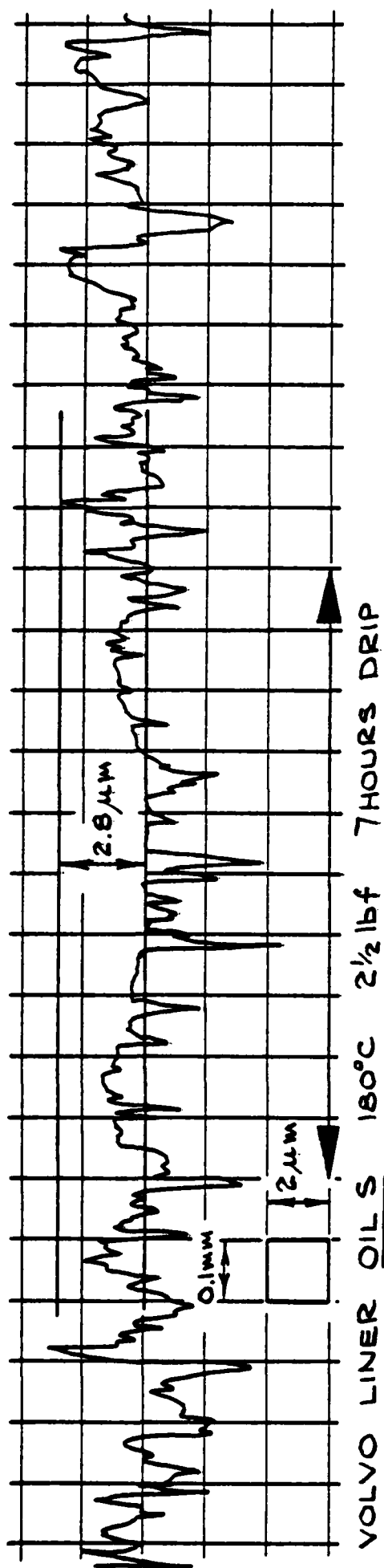
Measurement of Wear

A quantitative method of measurement was needed, so optical or scanning electron microscopy was excluded. The small wear scar, 1mm long, made surface roughness measurements complex, though they had been used earlier (ref. 2). The area of the wear scar was



LINER RADIUS  $R_L$  50 ~ 75 mm  
 RING RADIUS  $R_r$  RING A = 77/2 mm  
 $R_r$  ~ B = 81/2 ~

Fig. 5. Liner and Wear Scar



imprecise due to the absence of accurately defined boundaries.

The depth of the scar, measured by a Talysurf 4, was found to be the best method. The Talysurf has a 3mm travel so the skid can move 1mm each end over unworn material, so giving a datum line. The stylus of the Talysurf, observed through a microscope, was placed in the middle of the wear scar, and run axially along the liner. (Fig.5)

A typical trace of the wear scar and method of assessing the wear depth is shown in Figure 6. The depth is measured from the line through the peaks of the unworn surface to the bottom of the scar. This was simple for the US Army liner and the plateau honed liner K. It was harder for the other liners due to their larger roughnesses. However, with practice it was found that results could be obtained reproduceable to 5%.

## PART 2. CORRELATION WITH ENGINE TESTS

### Introduction

The percentage of cylinder bore polish is defined as the area of mirror surface ( $\sim 0.1\mu\text{m}$ ) formed by wear. The normal HFR wear scar is rougher than this, but a surface of this type can be obtained with a used crankcase oil containing fines as lubricant. The HFR may well show the start of the polishing process, suitable fines not being generated by the "once through" oil system. Good correlation is to be expected as the frictional systems in engine and test apparatus are matched in all important respects, the only exception being the combustion atmosphere, discussed later in this report.

### Wear Sequence 1

The Wear 1 sequence at 250°C did not give a clear distinction between the two reference oils RL47 and RL48. Now RL47 is the low polish oil and RL48 the high polish. It was considered that perhaps the temperature was too high so it was reduced to 180°C. It was found however that the high electrical contact resistance, a feature of low wear, failed after 30 minutes with RL47. Wear scar depth of  $2\mu\text{m}$  was found for RL47 as opposed to  $1.5\mu\text{m}$  for RL48.

The temperature was next reduced yet further to 160°C with the test conditions unchanged. Considerable degradation of the lubricant was observed. To avoid this, the lubricant was supplied by oil drip instead of by immersion.

### Wear Sequence 2. Drip Feed.

As soon as the oil was supplied by drip, oil RL47 gave a high ECR throughout the whole 3 hours of the test, at 160°C. There was a clear difference of wear between RL47 and RL48 as is shown below:

OIL	WEAR
RL 47	0 - $0.2\mu\text{m}$
RL 48	1 $\mu\text{m}$

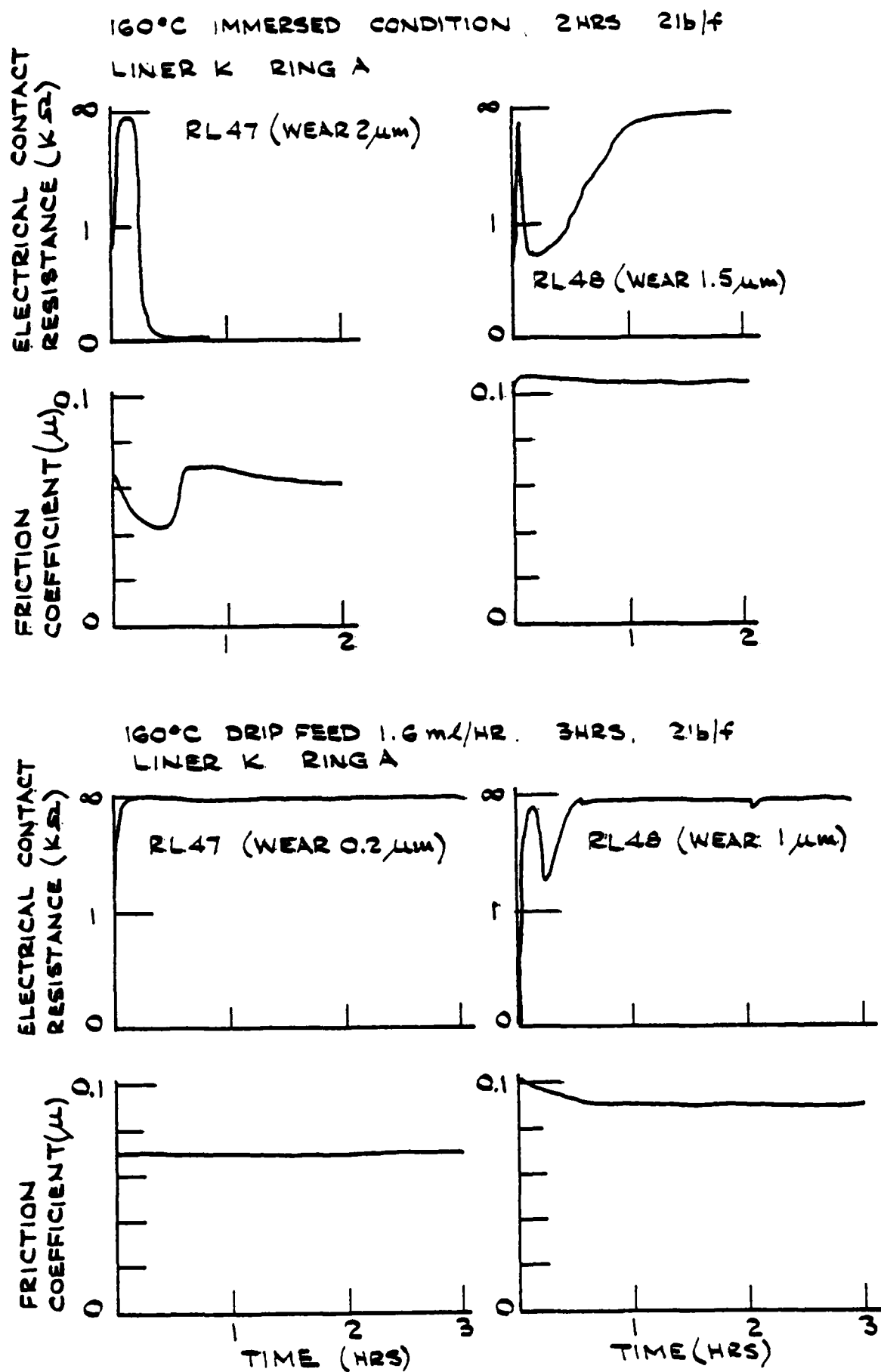
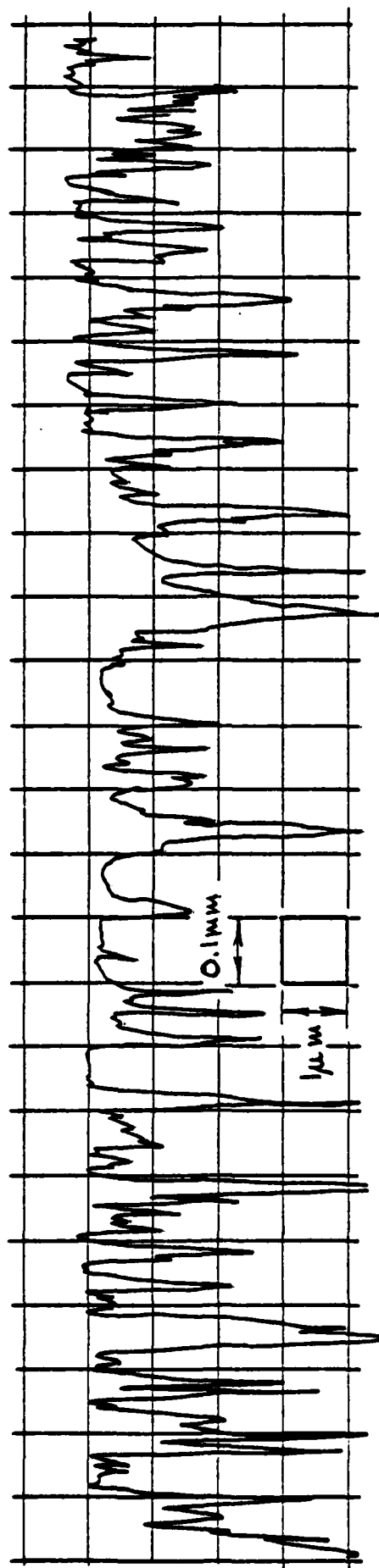
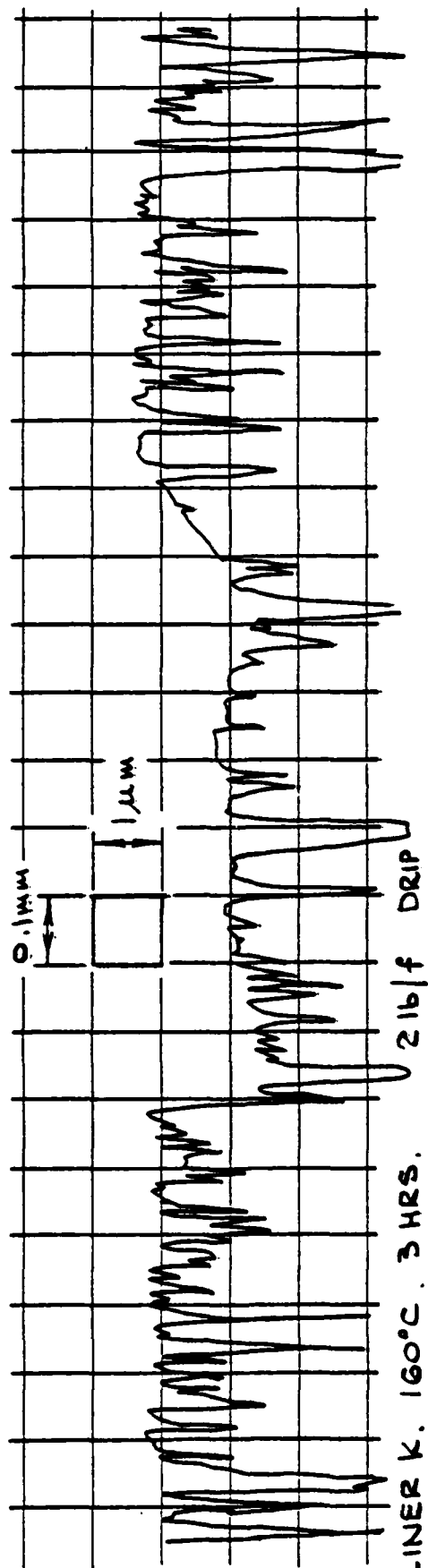


Fig. 7. Friction and Electrical Contact Resistance, ECR, RL47 & RL48,



RL 47. LINER K.  $160^{\circ}C$ . 3 HRS 21b/f DRIP  
RING. A.



RL 48. LINER K.  $160^{\circ}C$ . 3 HRS.  
RING. A.

21b/f DRIP

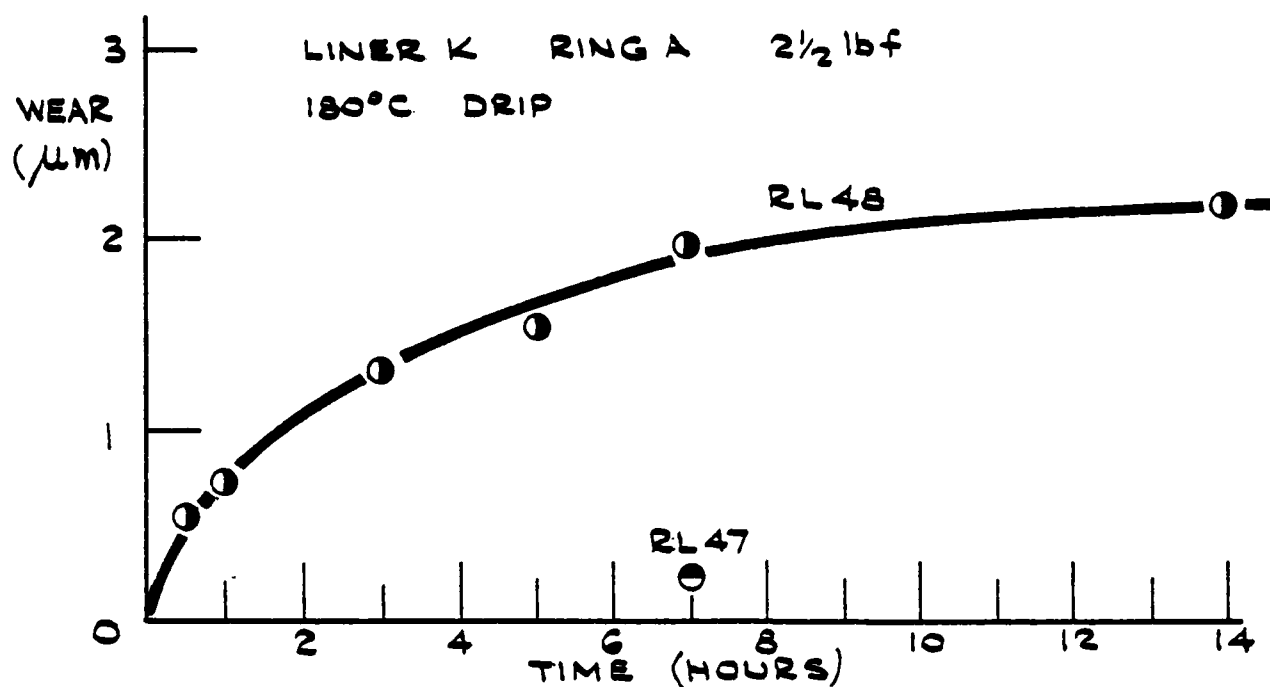


Fig. 9. Wear - time. RL47 & RL48. Wear 2 Liner K Load  $2\frac{1}{2}$  lbf 180°C Drip Feed

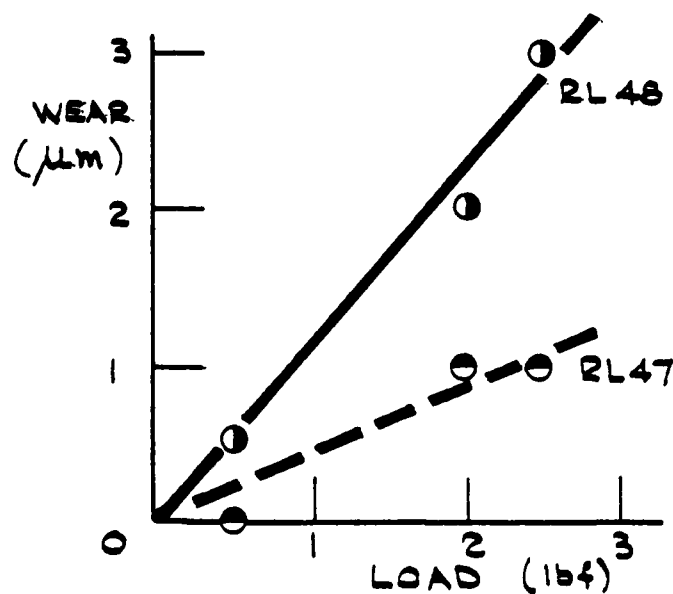


Fig. 10. Load - wear. RL47 & RL48. 7 hour test Wear 2 Tornado liner 180°C Drip Feed

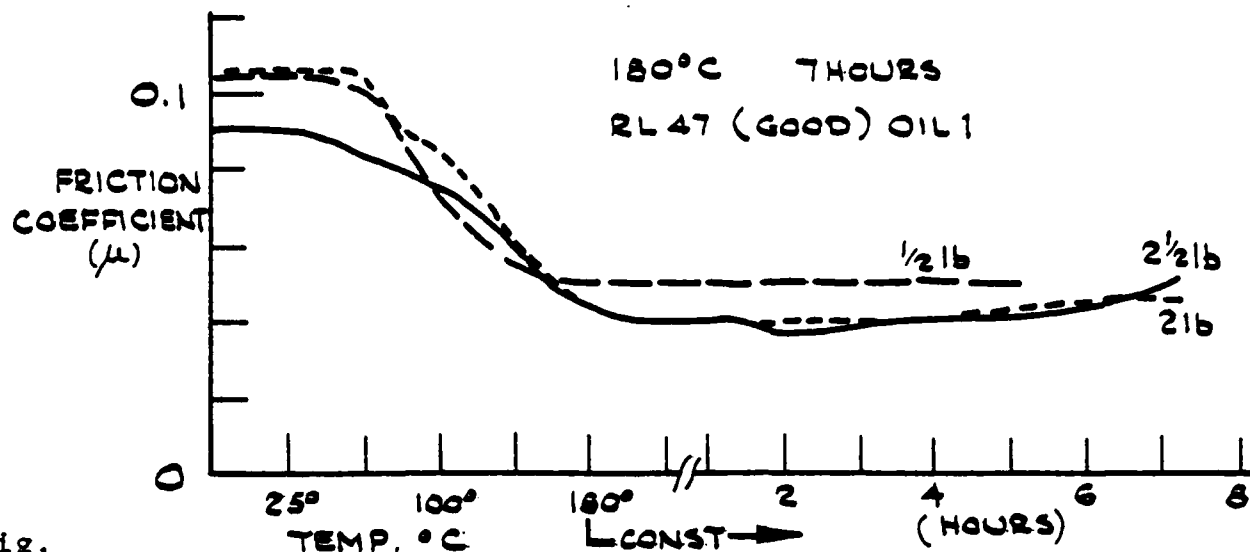


Fig. 11. Friction - time. Loads  $\frac{1}{2}$ , 2,  $2\frac{1}{2}$  lbf. Heating rate 7°C/min to 180°C. Tornado liner RL47 Wear 2 Test duration 7 hours. Ring A

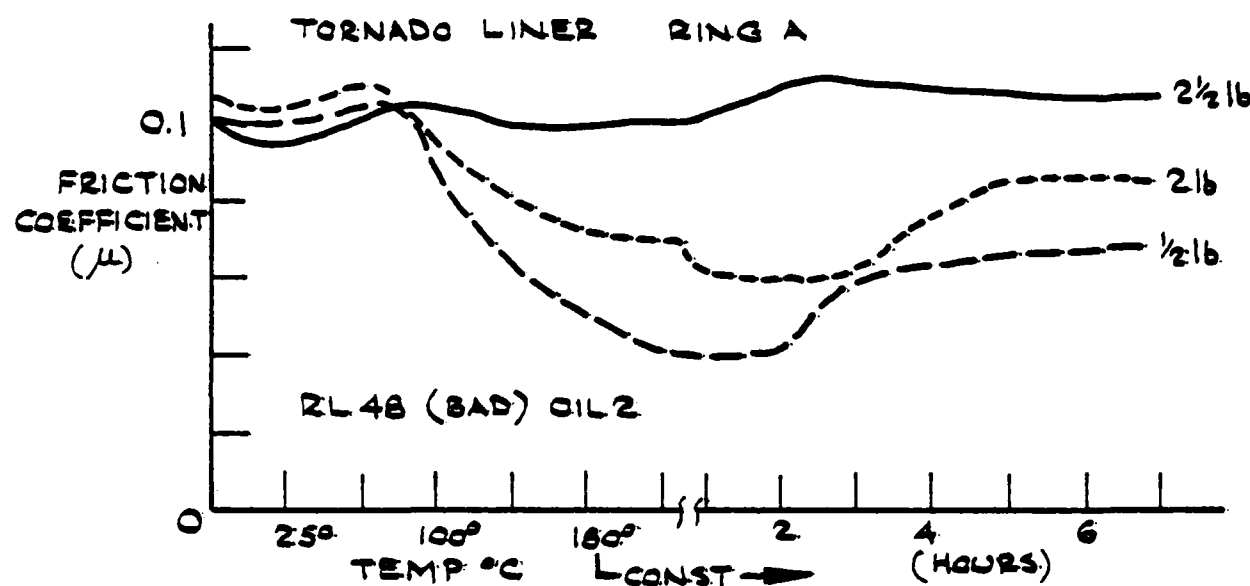


Fig. 12. Friction - time. Loads  $\frac{1}{2}$ , 2,  $2\frac{1}{2}$  lbf Tornado liner RL48 Drip feed Wear 2 Test duration 7 hours

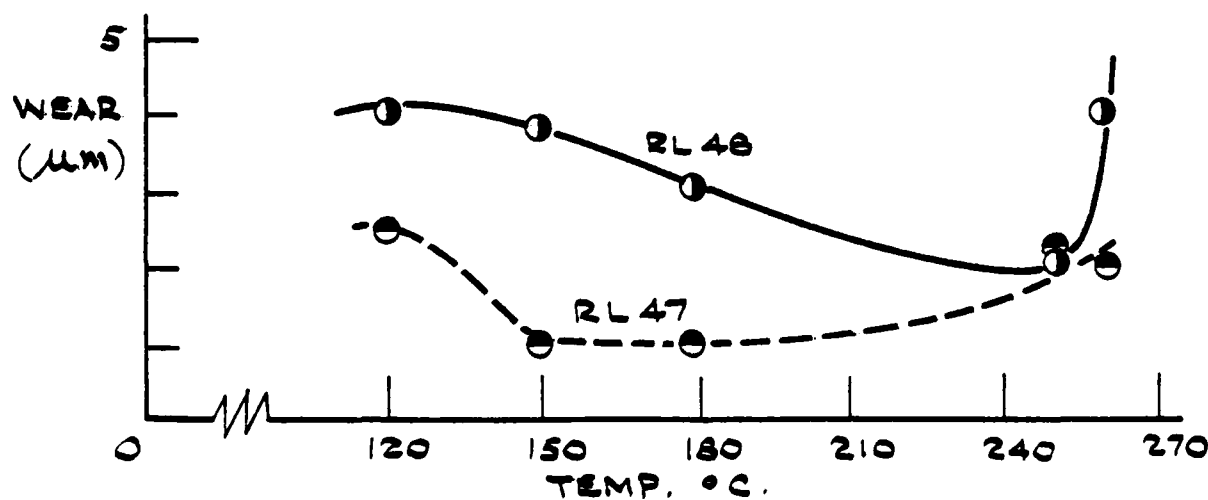
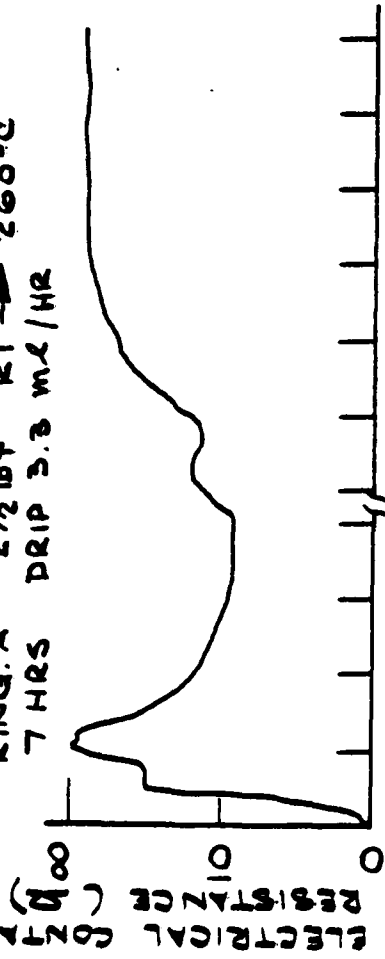


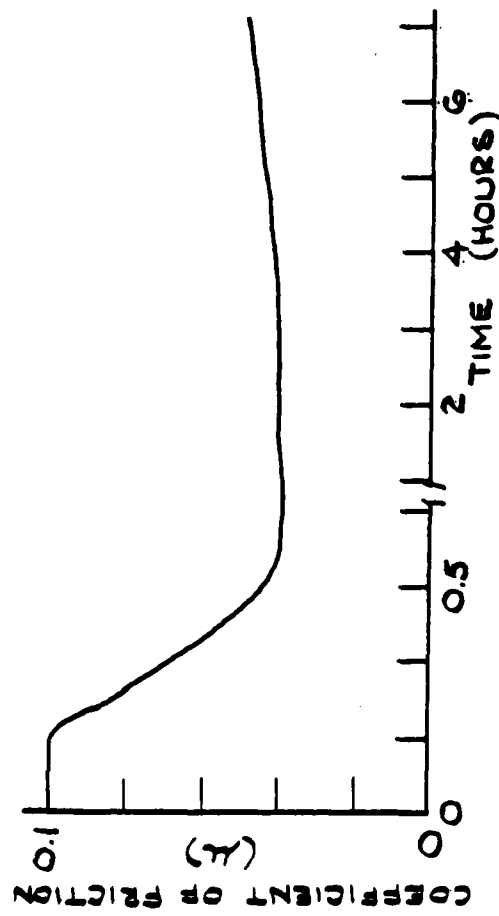
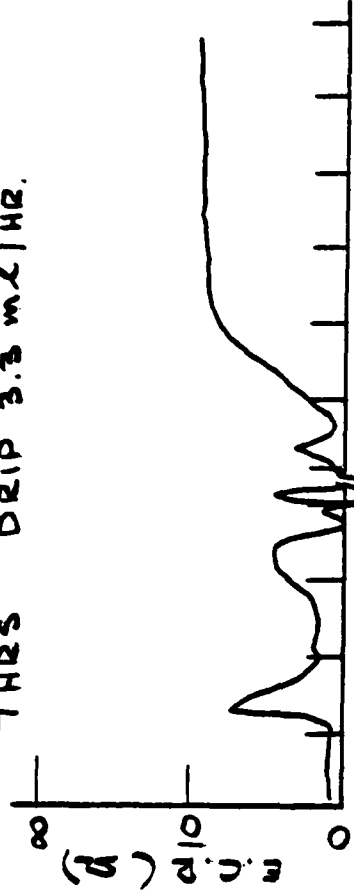
Fig. 13. Wear - temperature. Load  $2\frac{1}{2}$  lbf 7 hour test Tornado liner RL47 & RL48



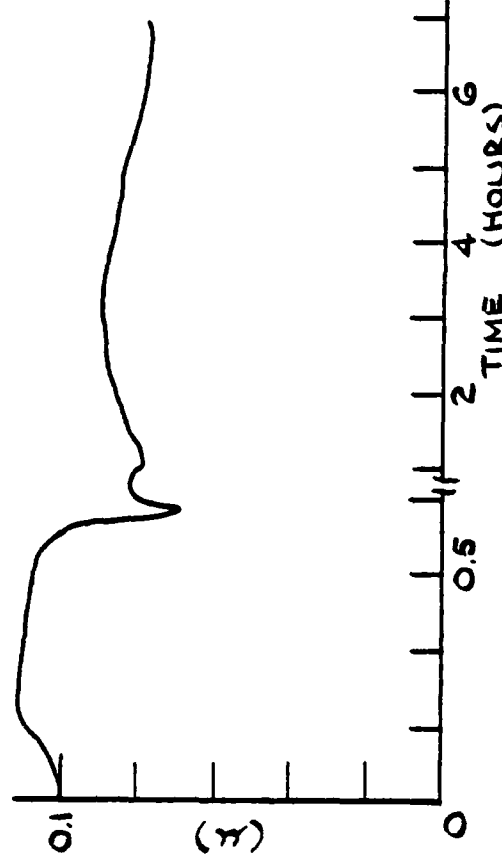
TORNADO LINER ; RL47 (WEAR 1 $\mu$ m)  
RING. A 2 1/2 lbf RT  $\rightarrow$  260 $^{\circ}$ C  
7 HRS DRIP 3.3 mL/HR



TORNADO LINER, RL48, (WEAR 4.0 $\mu$ m)  
RING A. 2 1/2 lbf RT  $\rightarrow$  260 $^{\circ}$ C  
7 HRS DRIP 3.3 mL/HR.



25 2560 130 200 260 CONST  $\rightarrow$  260  
TEMP $^{\circ}$ C



25 2560 130 200 260 CONST  $\rightarrow$  260  
TEMP $^{\circ}$ C

The friction and ECR results for the Wear 1 and Wear 2 sequence tests are shown in Figure 7 and the wear scar is given in Figure 8.

Tests in the zone where there was a high electrical contact resistance seemed promising. It was decided to choose a temperature of 180°C as standard for these wear tests for the following reasons. At temperatures of > 250°C the scar showed score marks, which were not present in polished engine liners. While at temperatures above 200°C heavy deposits are formed, which make wear volume determinations difficult.

In general, high temperatures are desirable as they produce large wear, so giving short tests. It was considered a value of 180°C was the highest that could be used without incurring excessive oil varnish or degradation.

Another point is that the lower temperature, 180°C, probably represents mid-stroke conditions. The higher value, 250°C is closer to top dead centre. The wear tests showed that 180°C gave greater discrimination between the oils than 250°C. Several authors (4,5) have suggested that 180°C is a good approximation to mid stroke conditions.

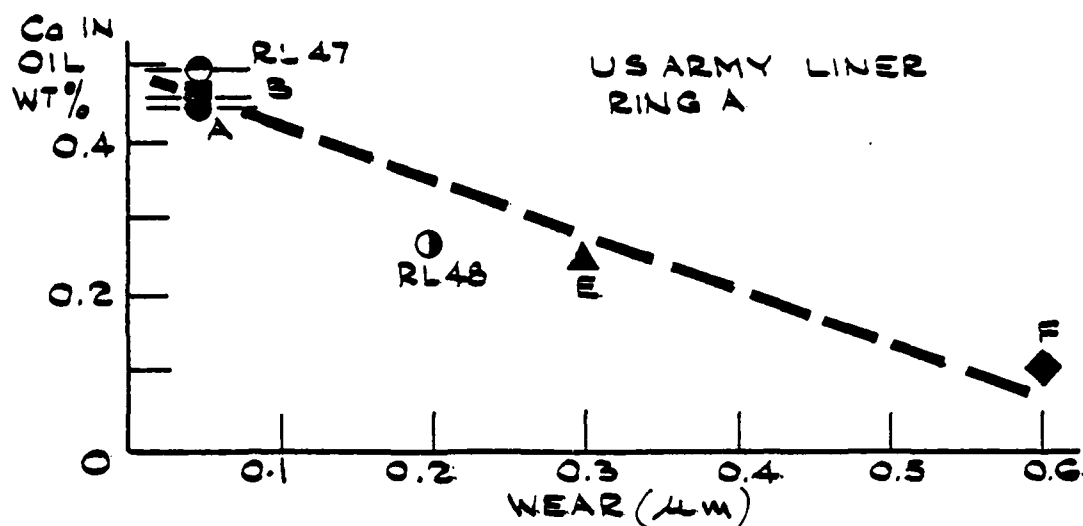
In Figure 9 some wear results are shown which were taken to determine how long a test was necessary. It is seen that wear almost stops after 7 hours. A surprising finding was how differently the various liners responded to the same oils. Also load was found to be important for good discrimination as the results which are listed below, and are plotted in Figure 10, show.

Load lbf	WEAR $\mu$ m	
	RL 47	RL 48
0.5	nil	0.5
2.0	1	2
2.5	1	3

They give a roughly linear relation between load and wear for the two oils RL47 and RL48.

The friction traces as a function of time for these two are shown in Figures 11 and 12.

A further series of tests was done to confirm the choice of 180°C as the decisive temperature. Two temperature zones were chosen. (Fig.13) In the one, of 180°C and below, a high electrical contact resistance (ECR) was normally present. In the other 250°C and above, the ECR was usually almost zero. Typical friction and ECR traces are in Figure 14. In these traces the temperature reaches 260°C over 30 minutes at which it remains for the rest of the 7 hour test. In the low wear oil, RL47, the ECR is high and the friction low. RL48 gives much lower ECR, but higher friction and wear.



5. Calcium in oil wt% - Wear. US Army oils and RL47 & RL48 US Army liner  
Wear 2 7 hour test. Ring A

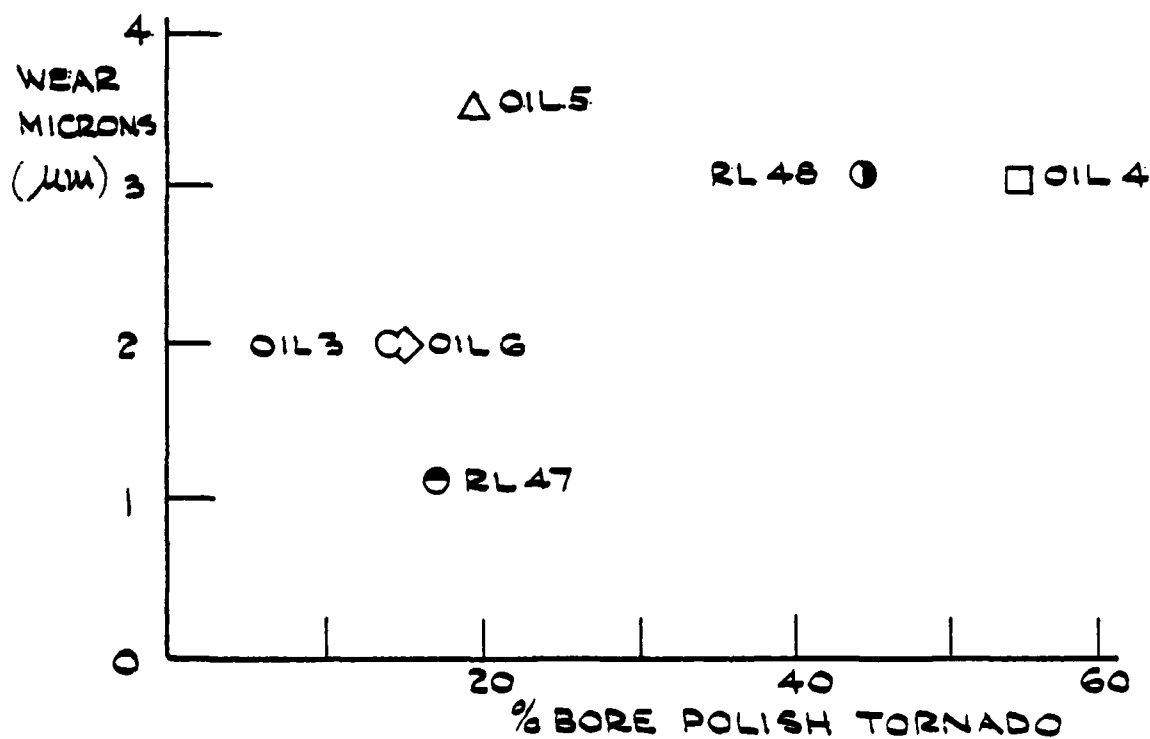


Fig. 16. % Bore Polish - wear. Tornado liner. 6 Mobil oils. Wear sequence 2

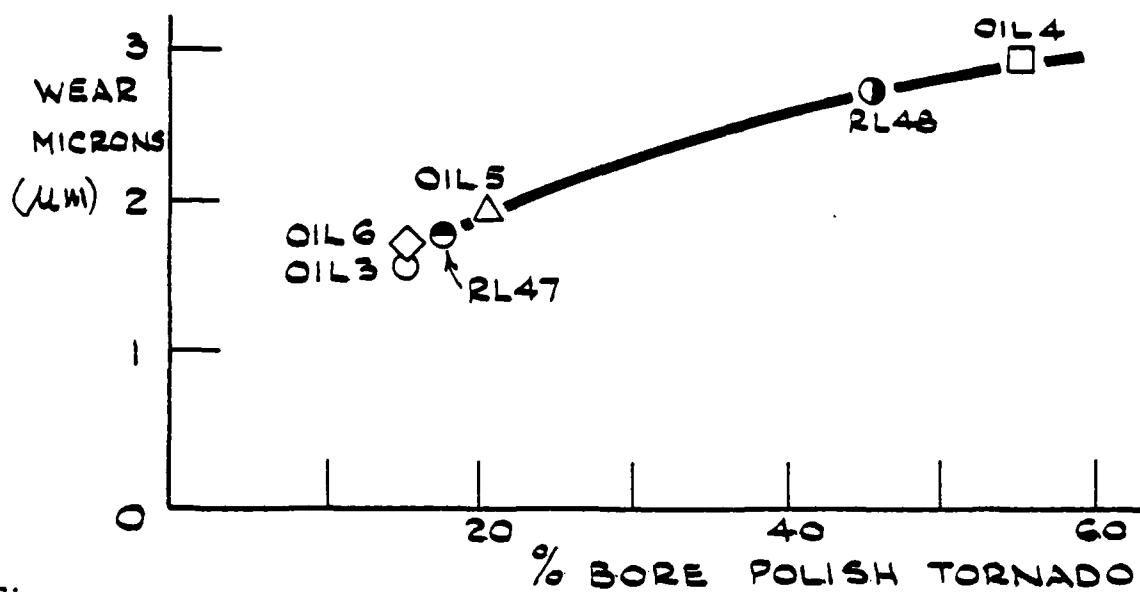
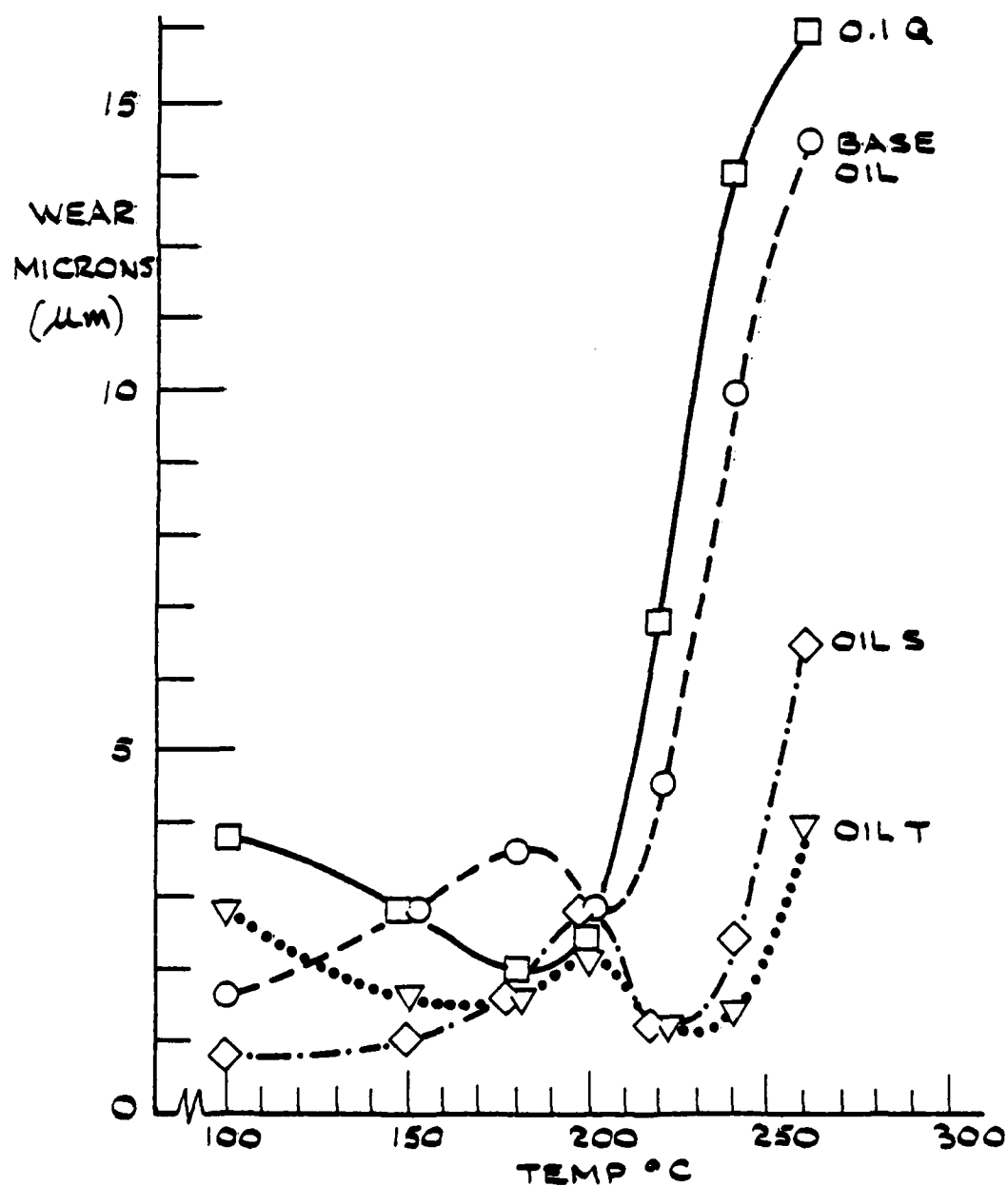


Fig. 17. % Bore Polish - wear. Tornado liner. 6 Mobil oils. Wear sequence 3



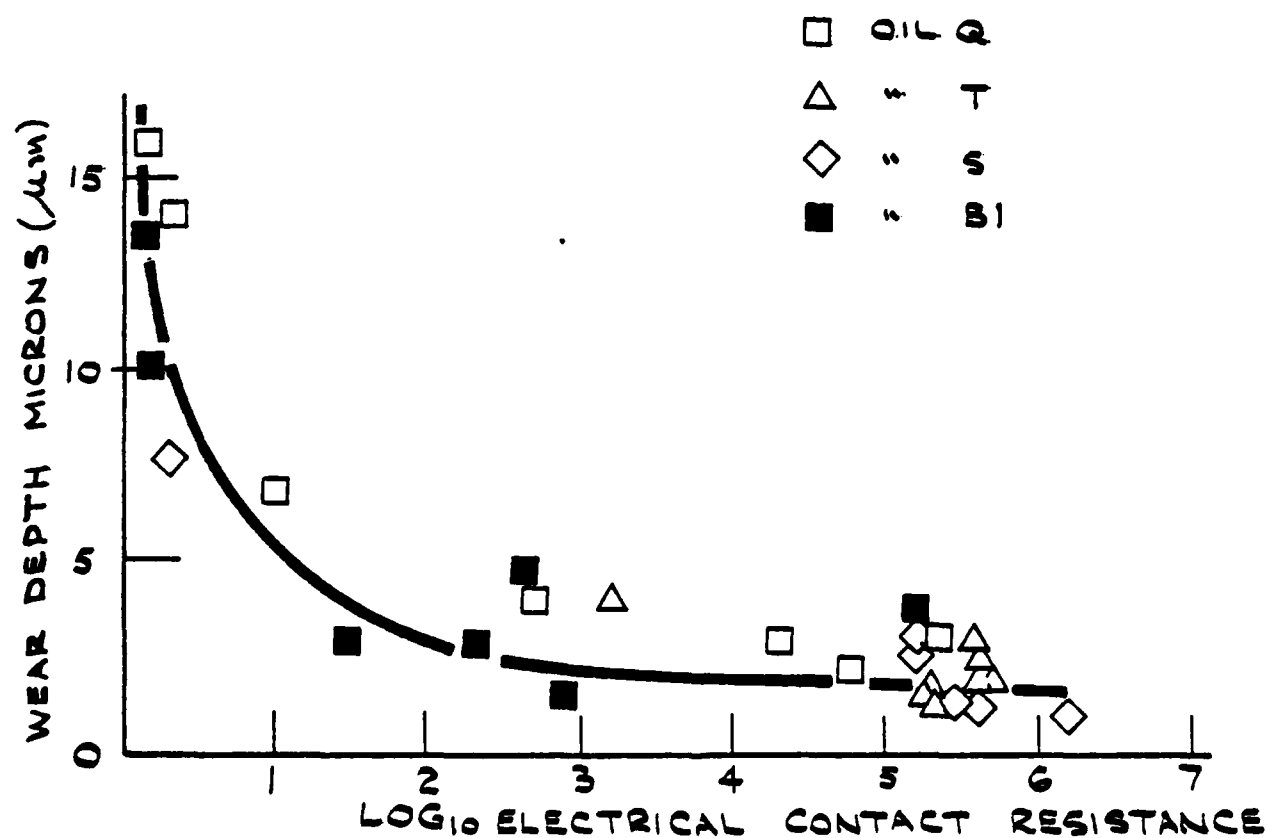
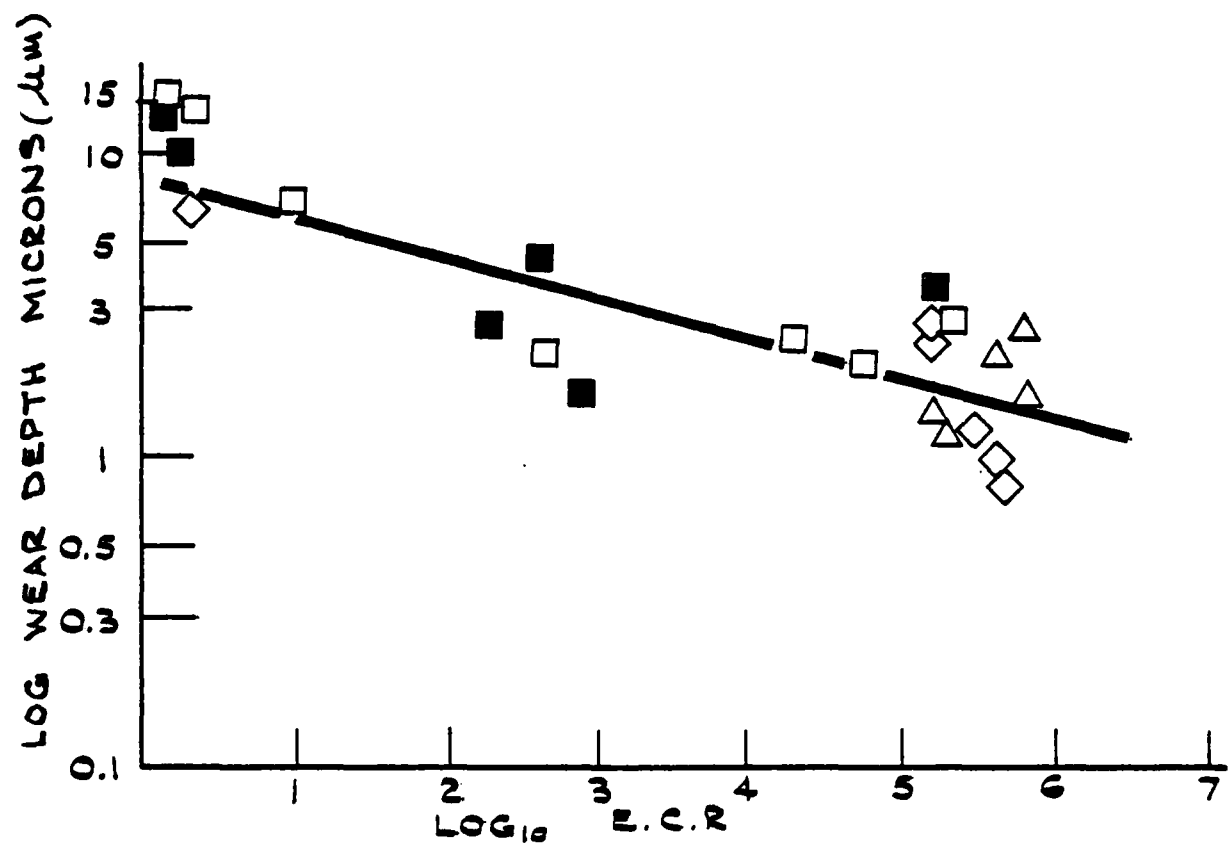
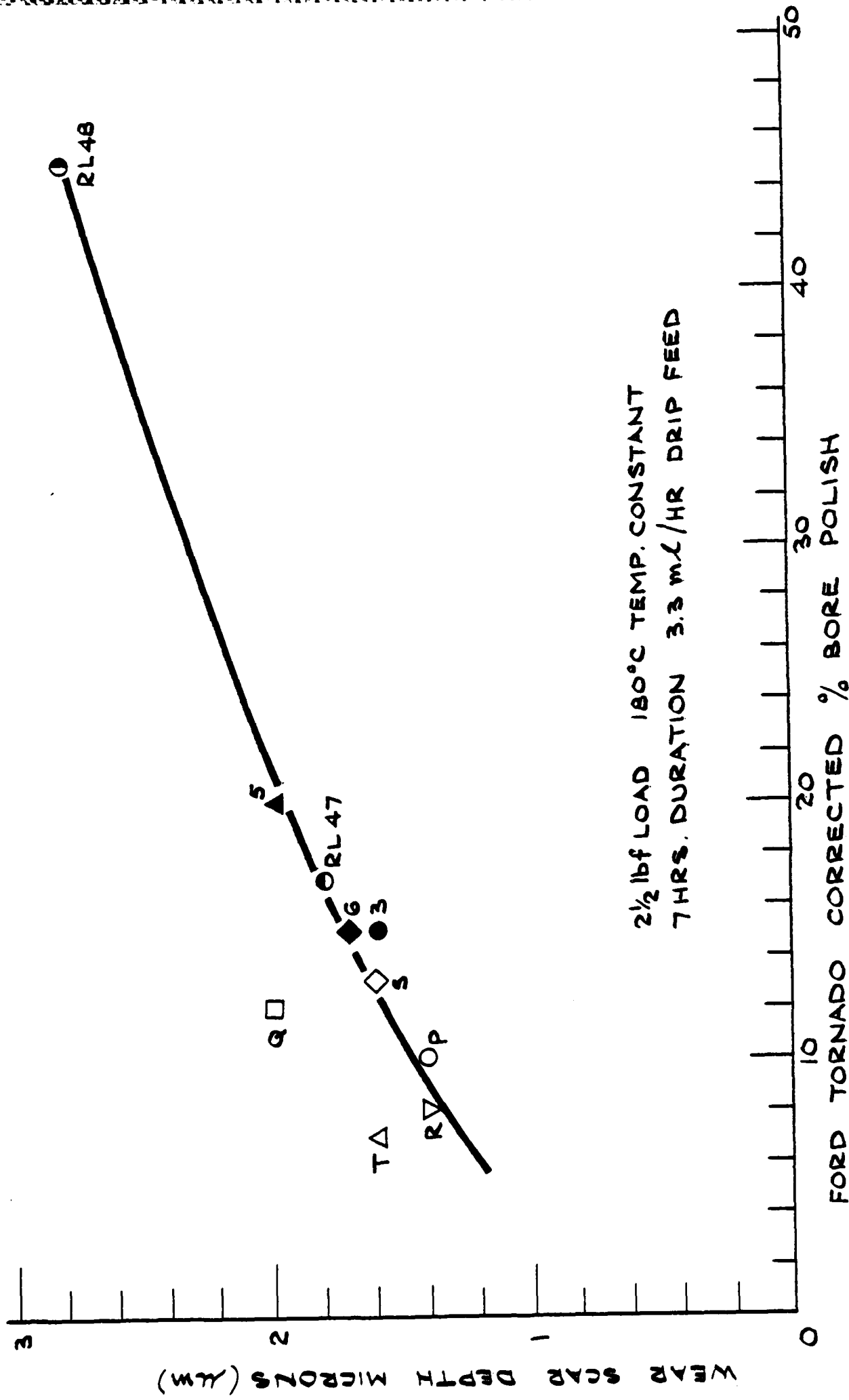


Fig. 19. Log Electrical Resistance - Wear depth from Fig.18.



During these tests it was noted that the amount of calcium detergent in the oil markedly affected the wear. This finding, which will be discussed later, is illustrated in Figure 15.

### Wear Sequence 3

The 6 oils C1 to C6, which were provided by Mobil Ltd of London, were next tested. Oil C1 is RL47 and C2 is RL48. These were first run under the Wear 2 sequence and the correlation with bore polish (as obtained in the 200 hour Tornado test) was poor, as Figure 16 shows. Oil 5 was especially poor and it was noticed that there had been rapid wear at temperatures much lower than the 180°C test temperature.

The apparatus was therefore run up to operating temperature of 180°C with the specimens out of contact. There was a dramatic improvement in the correlation, as Figure 17 shows. This test sequence is Wear 3 and became the standard.

### Importance of Liner Material

Next the 5 Castrol oils, which had been rated in both Tornado and Volvo tests, were studied. The Tornado had rated these five better than the good reference oil RL47. In the Volvo, however, three were shown to be as bad as RL48. First it was suspected that the Volvo had a higher running temperature than the Tornado.

Tests were therefore run at 260°C using a Tornado liner. The results were inconclusive.

A detailed study of three of the oils was therefore made. S, which had performed best in the Volvo, and Q and T, which were worst, were chosen as well as their base oil. The results for wear at temperatures between 100°C and 260°C are shown in Figure 18.

In Figure 19 the electrical contact resistance ECR (logarithmic values) are plotted against wear. The bottom curve gives wear against log resistance. In the top, log wear depth is plotted against log electrical resistance. It is interesting to note that this is a straight line. Despite the warning of Fiennes and Anderson, Ref.(6), it would be an attractive hypothesis to suggest that log resistance is a measure of film thickness and these two graphs indicate how wear increases as the film reduces in thickness. Again more work is needed here.

The correlation between these oils and the Tornado engine results is good, as Figure 20 shows. The Tornado liner was used for the HFR tests under the standard Wear 3 conditions.

Next, the effect the liner was investigated. It was not suspected that the oils would respond very differently to different commercial liners. It was found, however, that the liner did make a considerable difference. Using the Volvo liner the correlation became excellent, as Figure 21 shows. This emphasized the importance of using the correct liner.

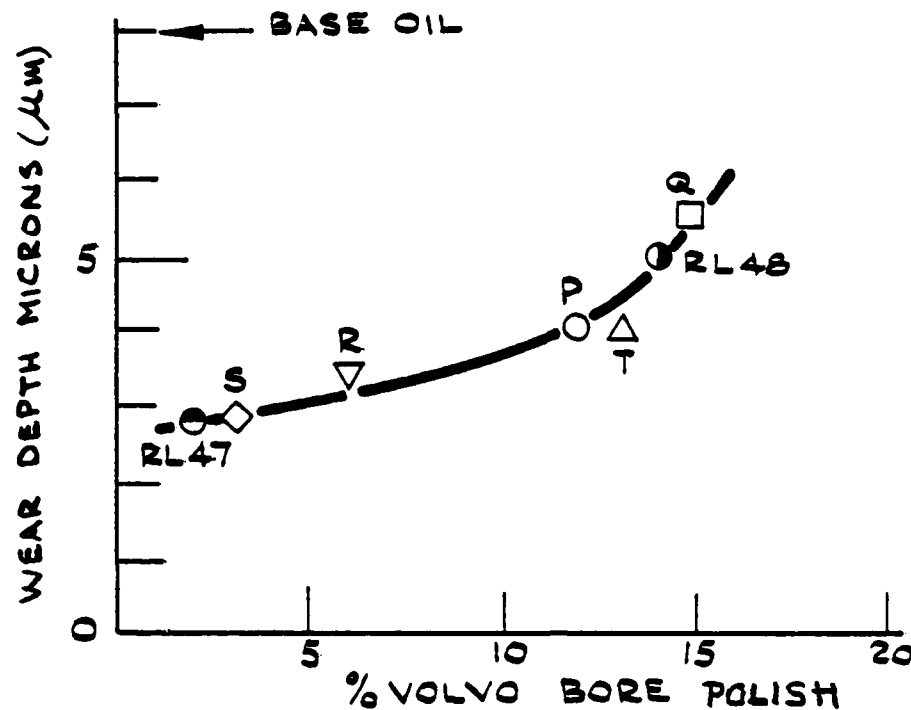


Fig.21. Wear - Volvo Bore Polish. Volvo liner.

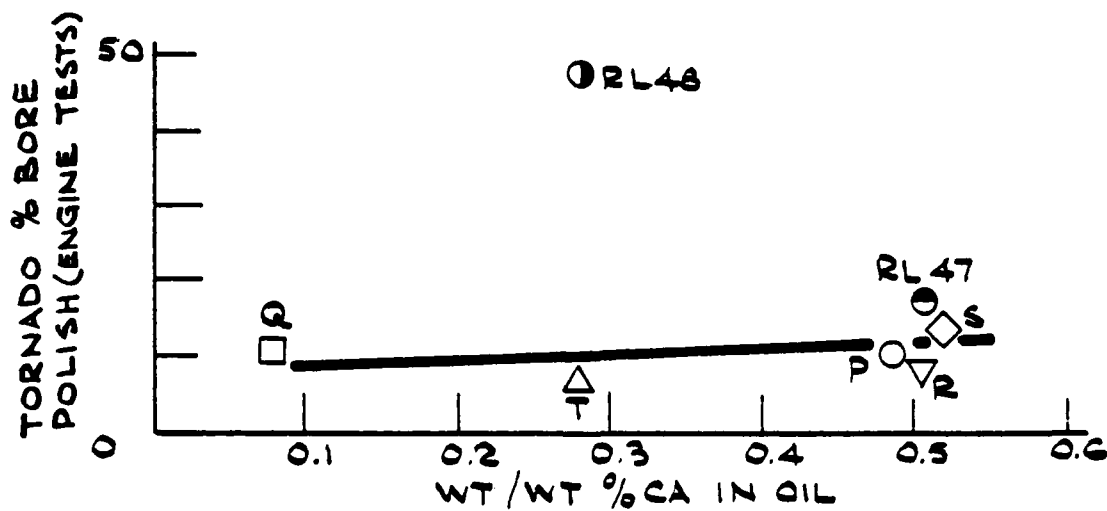
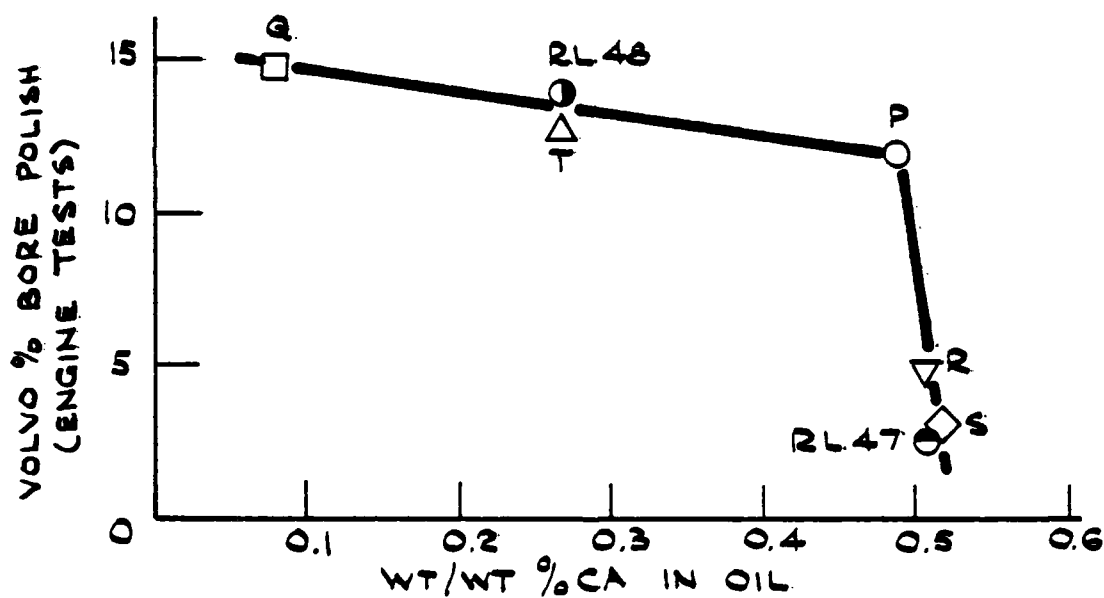


Fig. 22. Calcium in oil - Bore polish. a) Volvo b) Tornado



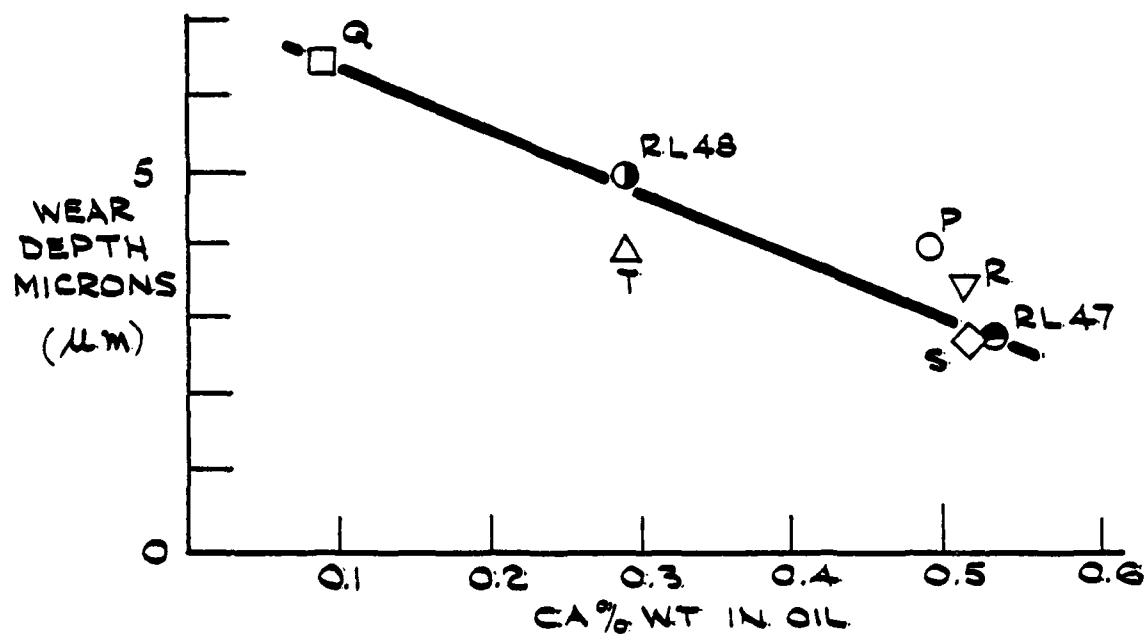


Fig.23. Calcium in oil - wear depth. Cross plot of Figs.21 & 22 a)

Also, the amount of calcium detergent in the various oils was plotted against the percentage Volvo bore polish in Figure 22a and for the Ford Tornado in Figure 22b, an unexpected difference. In Figure 23 a cross plot of Figures 21 and 22a gives wear against calcium content. These results confirm the wear data shown earlier (Figure 15) for the effect of calcium on wear for the US Army oils.

#### 6V 53T Scuffing Test

##### a) Wear

Next, having obtained considerable experience in wear tests the influence of wear on scuffing was studied. It is usually assumed that there is no correlation between the two phenomena. The scuffing data given in US Army Interim Report 112 was compared with the 7 hour wear results. This is shown in the table below.

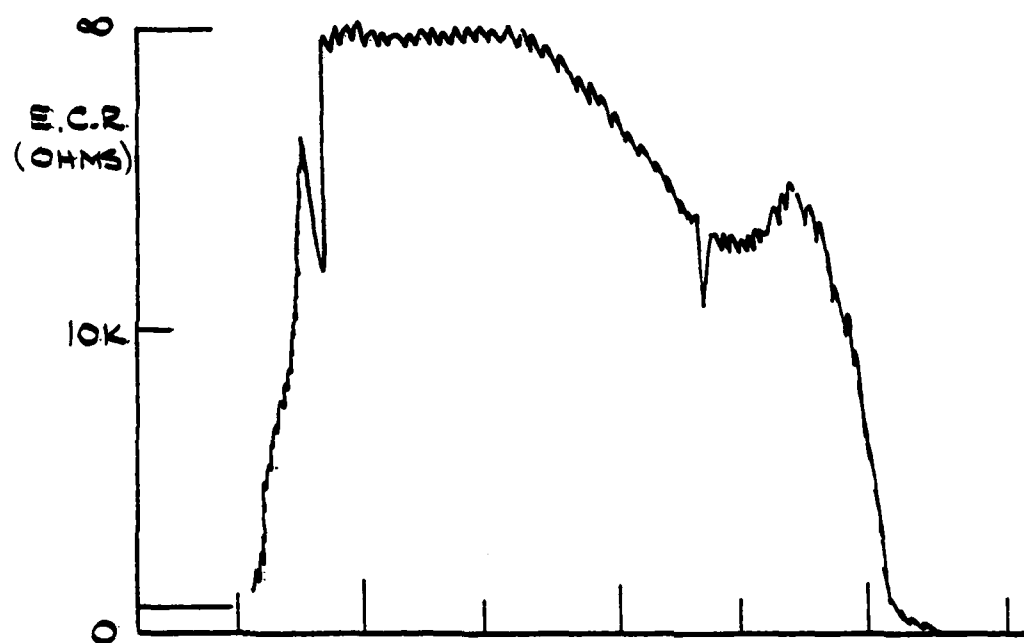
Oil Designation	HFR Wear 7 hours 180°C µm	% Liner Area Scuffed 240 hour US 6V 53T Test
AL 6856L A	0.1	18
AL 6855L B	0.1	38
AL 8924L E	0.3	9*
AL 7172L F	0.6	11
AL10153L	0.5	not given
AL10722L	0.2	not given
RL47 (Oil No.1)	0.1	-
RL48 (Oil No.2)	0.2	-

\* In the rating data sheet C 1-7 of August 1979, of US Report 112, one cylinder of the engine, 2 Left, scuffed 85%. All the other 5 scuffed only 9% of the cylinder surface. If 2 Left is included, the area scuffed rises to 22%. It is omitted in this table.

Looking at the table for the 4 oils, ABEF, for which scuffing data is available, there appears to be no correlation between wear and scuffing. This was expected, and confirms that wear and scuffing are indeed separate phenomena as far as this system is concerned.

##### b) High Temperature

It was thought that the effective temperatures of the US Army 2-stroke engines were much higher than 180°C, which was correct for bore polishing, as shown above. The test procedure was therefore



3°C / MIN. TEMP. RISE. LOAD 316 f

U.S. ARMY LINER AL 6855L RING A

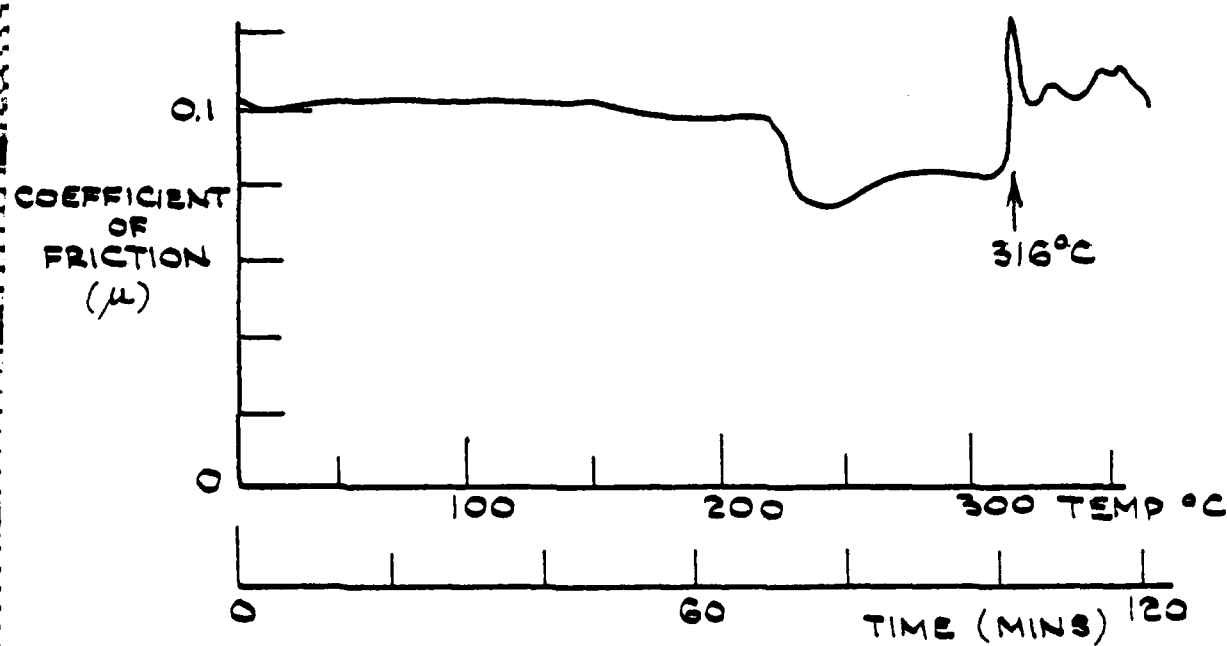


Fig. 24. Friction and ECR - temperature US Army liner Oil AL 6855L Ring A

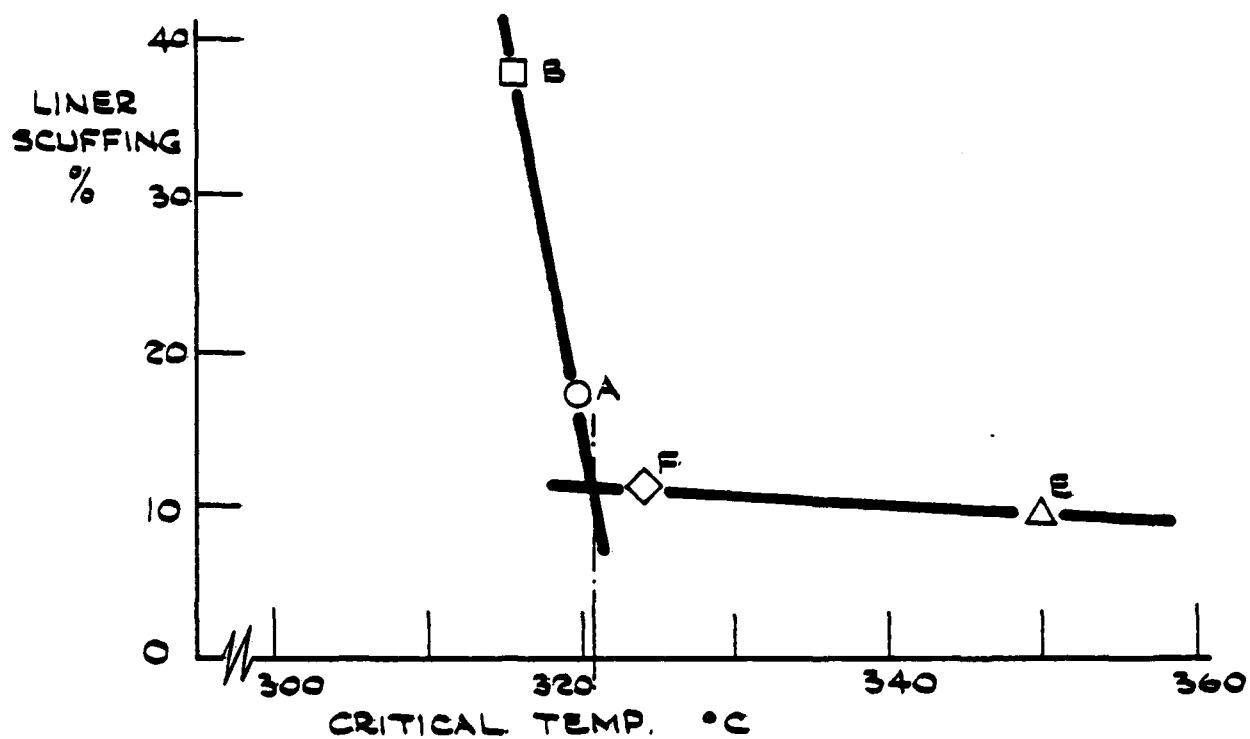


Fig.25. Friction and ECR - Temperature US Army liner Oil AL 6855L Ring A

altered. The friction and electrical resistance were monitored up to 360°C. The heating rate was lowered to 3°C/minute so the test took 2 hours. Drip feed at 3.3 ml/hour was again used.

A run with US oil B (AL6855L) under 3lb load is shown in Figure 24. The friction dropped at about 220°C. Most noticeable, however, is the very sharp rise which occurred at 316°C. It was considered that this very sharp rise indicated scuffing, followed by partial healing.

All 4 oils were tested at this load and the temperature at which the sudden jump took place was noted. The "critical" temperatures of the 4 oils, together with % scuffed area from the 6V53T test is listed below.

---

Oil	Critical Temp.°C	% Surface Scuffed
	3lb	6V53T. 240 hour
A	320°	18
B	316°	38
E	350°	9
F	324°	11

---

These results were plotted in Figure 25. The critical temperature, indicated by the HFR, is quite clearly related to the amount of scuffing in the 6V 53T test.

This is considered a notable finding.

#### c) Effect of Load on Scuffing

In the June 1983 interim report it was reported that load affected the scuffing temperature. This was later found to be due to an incorrect positioning of the thermocouple leading to an erroneous temperature reading. The reason for this error is that the contact point was "immersed" in a meniscus or drop of oil near the contact, and the temperature of the drop was found to be 10°C lower than the surface outside it. This 10°C difference was serious and emphasizes the critical importance of temperature and its accurate measurement.

Loads from ½lbf to 6lbf were tested, and the scuffing temperature was found to be constant. This is discussed later.

### PART 3. PROPERTIES OF THE HIGH ELECTRICAL CONTACT RESISTANCE ZONE

#### Introduction

The correlation between engine lubrication and the HFR rig has been established in part 2 of this report. In this section the nature of the high electrical resistance zone is studied. As far as is known no work has been done on this area of lubrication, as no test apparatus was available capable of studying it.

This part is divided into five sections.

- 1) lubricants
- 2) calcium detergent concentration
- 3) temperature
- 4) the effect of load
- 5) liner metallurgy and finish

Preliminary work showed that if a high contact resistance film is present, there is almost always low wear. In this context "high" is anything over 10 ohms. The resistance is usually far higher than this, as will be seen later. The friction coefficient is about 0.1. This region of high electrical contact resistance, or ECR, will be designated the Anti-wear, AW, region.

When the film is not there, indicated by a low ECR, often the friction is not very markedly changed, though the wear is usually greater. This zone, where the ECR is low, will be called the extreme pressure, or EP, region.

#### Lubricants

##### a) Additive Response.

The first observation on the effect of lubricants on wear was that the bulk viscosity is not the factor that decides if wear will take place. All the five oils, P to T, were all of the same viscosity grade SAE 15W/40. but showed quite different wear characteristics. The two US Army oils, AL-6855-L and AL-6856-L, had different viscosities but the same additive package, and so gave very similar wear results. The 6855 was a 10W (OE/HDO 10) and the 6856 a 30(OE/HDO 30).

The difference in wear is a function of the electrical resistance (ECR). Now Anderson and Fiennes (6) have shown very clearly that electrical resistance is not simply a function of oil film thickness, nor a quantitative measure of it. It does, however, give a qualitative indication of its size. This is demonstrated in Figure 18 where the wear is plotted against the contact resistance measured by the technique described earlier for two series of oils.

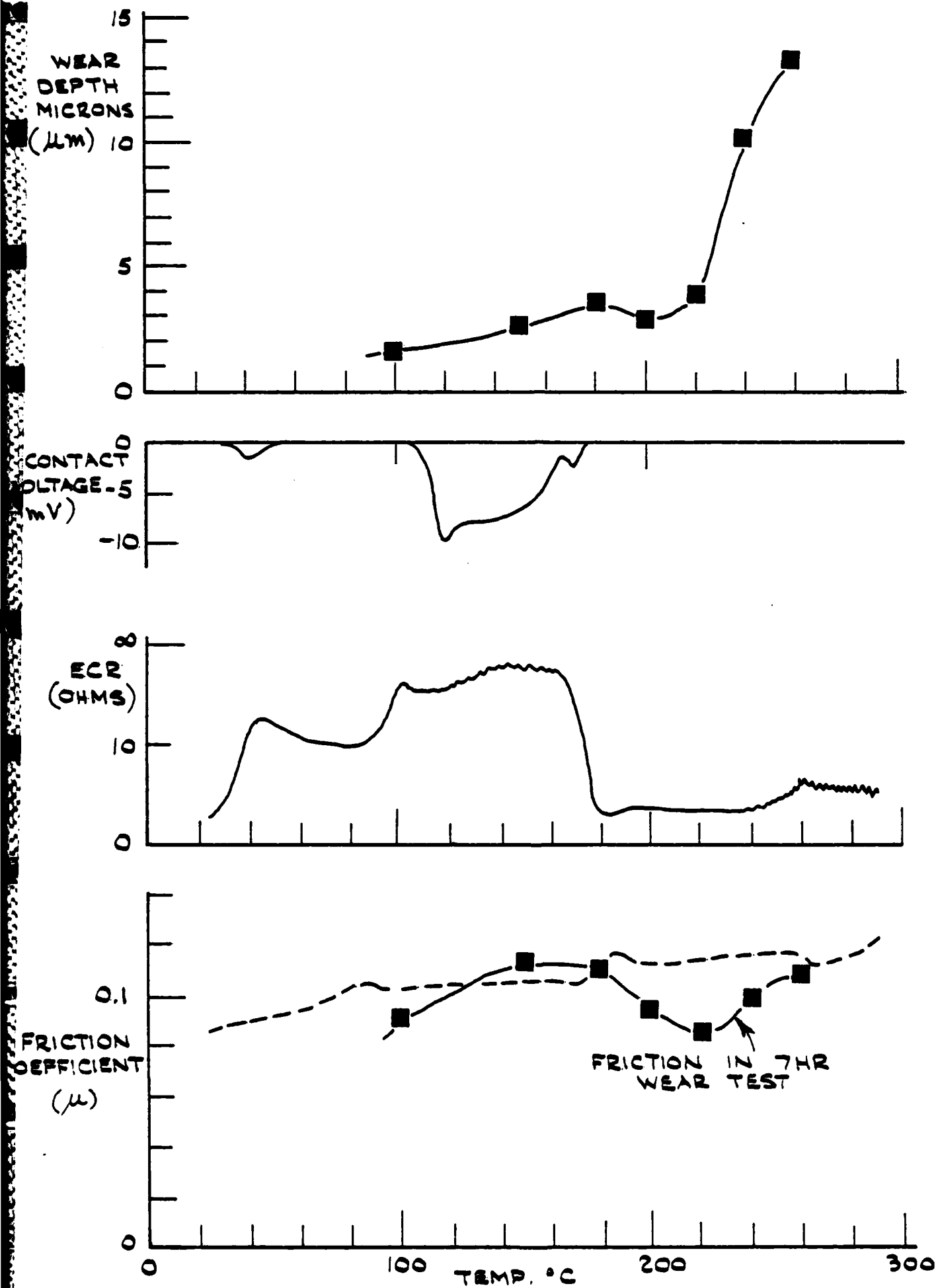


Fig. 26. US Army Scuffing Test 6V 53T - Critical Temperature Oils A B E F

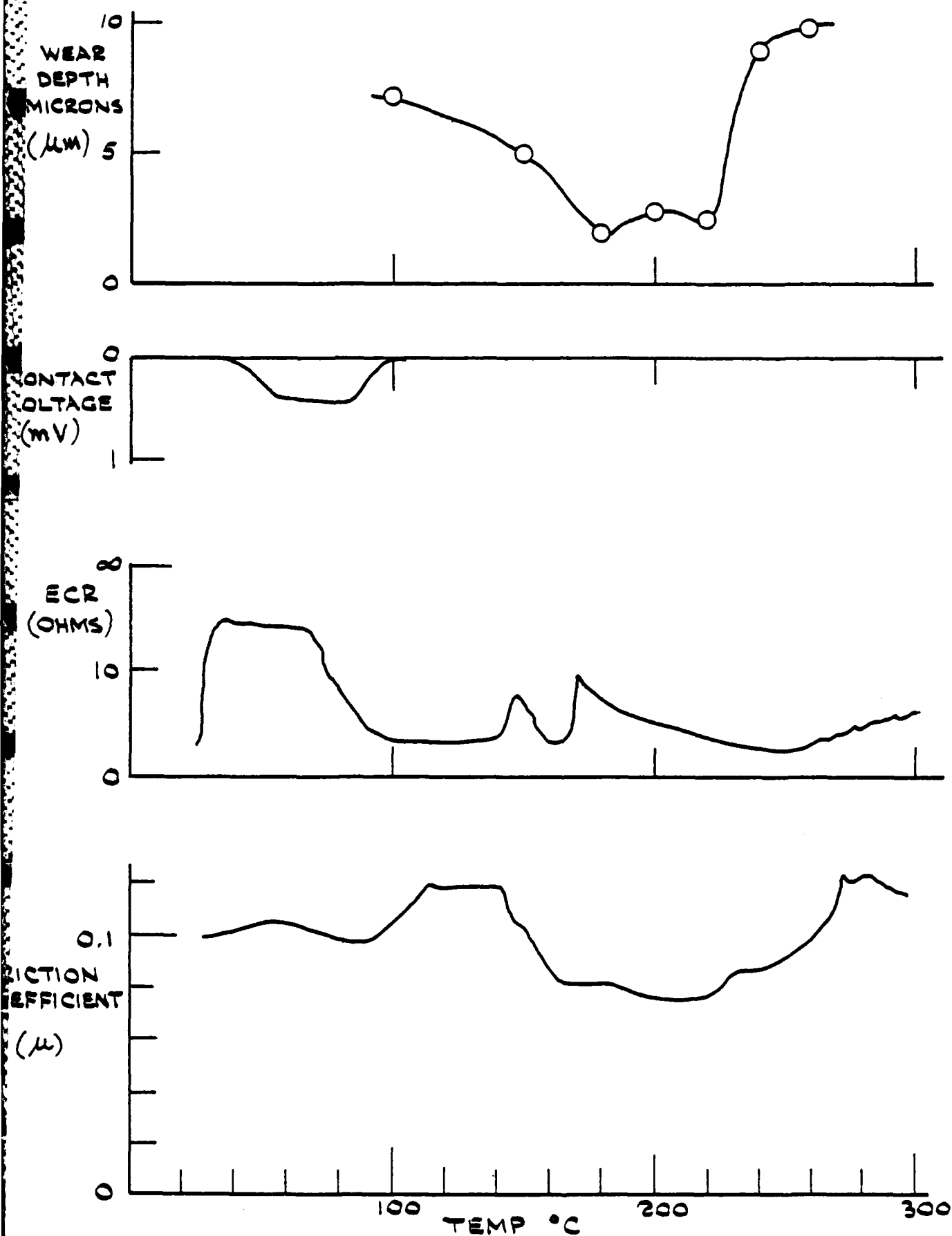


Fig.27. Friction ECR Contact Voltage - Temperature 350 NS Tornado liner Ring B



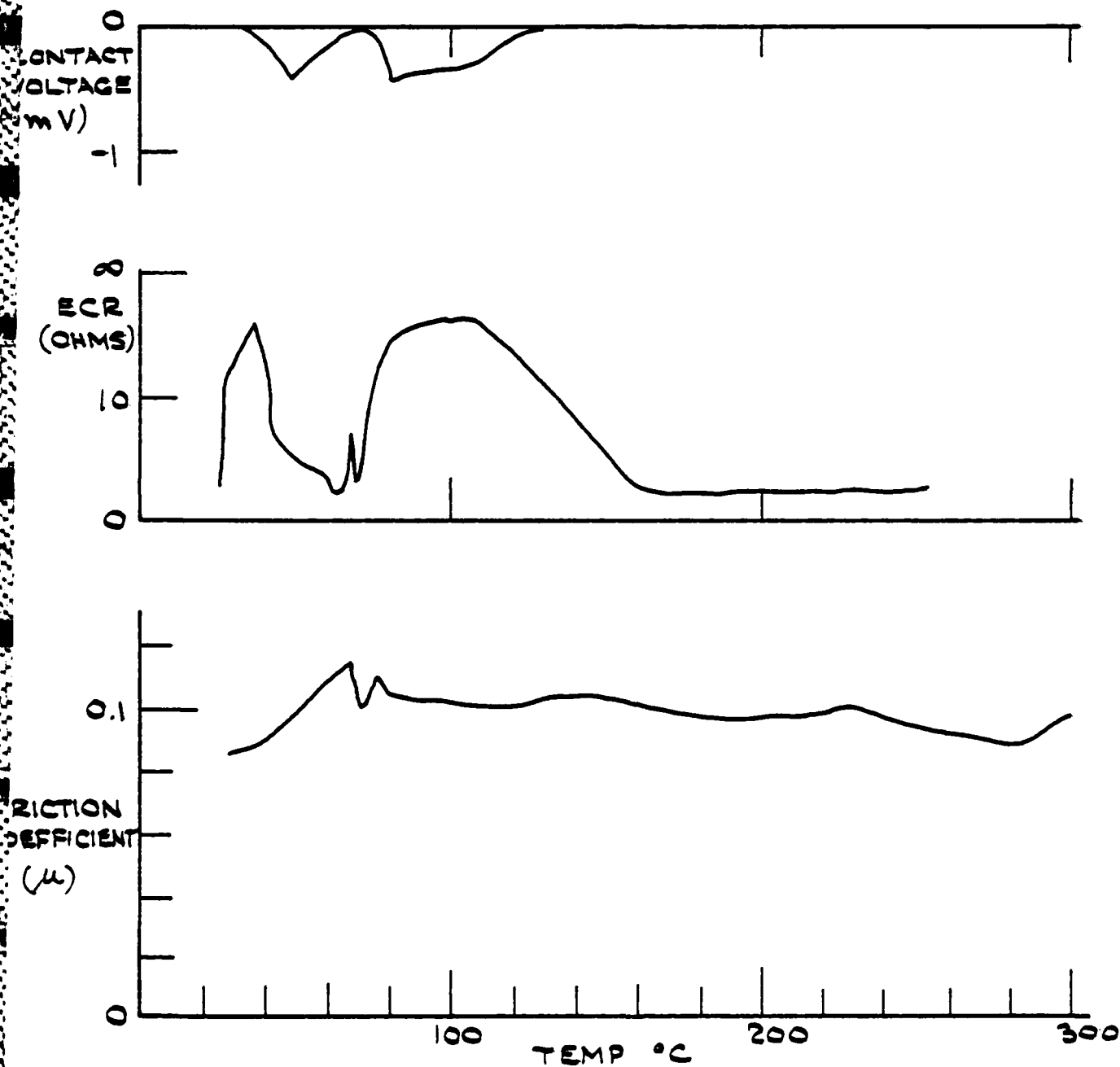


Fig.28. Friction ECR Contact Voltage - Temperature Purified 350 NS  
Tornado liner

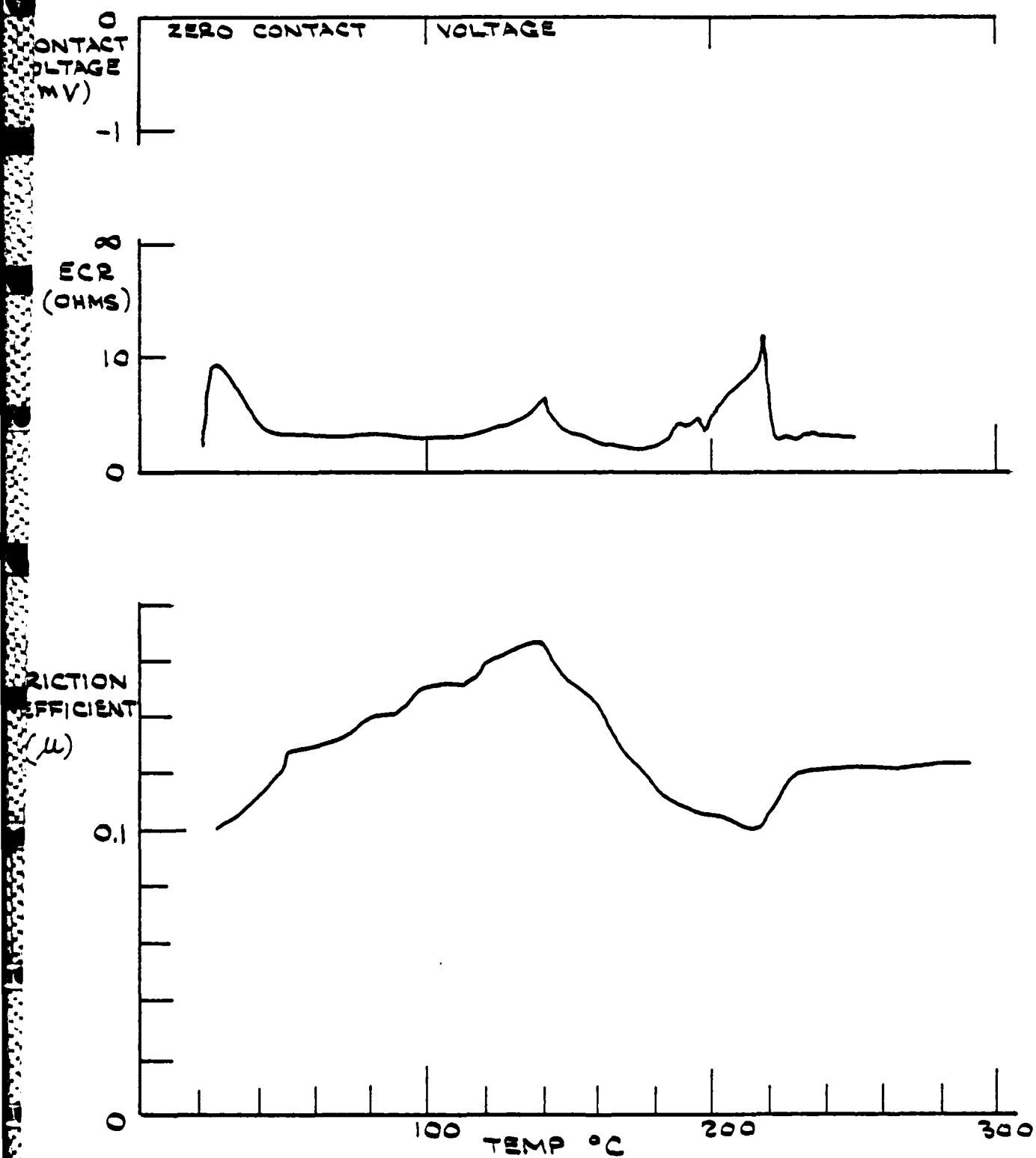


Fig.29. Friction ECR Contact Voltage - Temperature Synthetic hydrocarbon Tornado liner

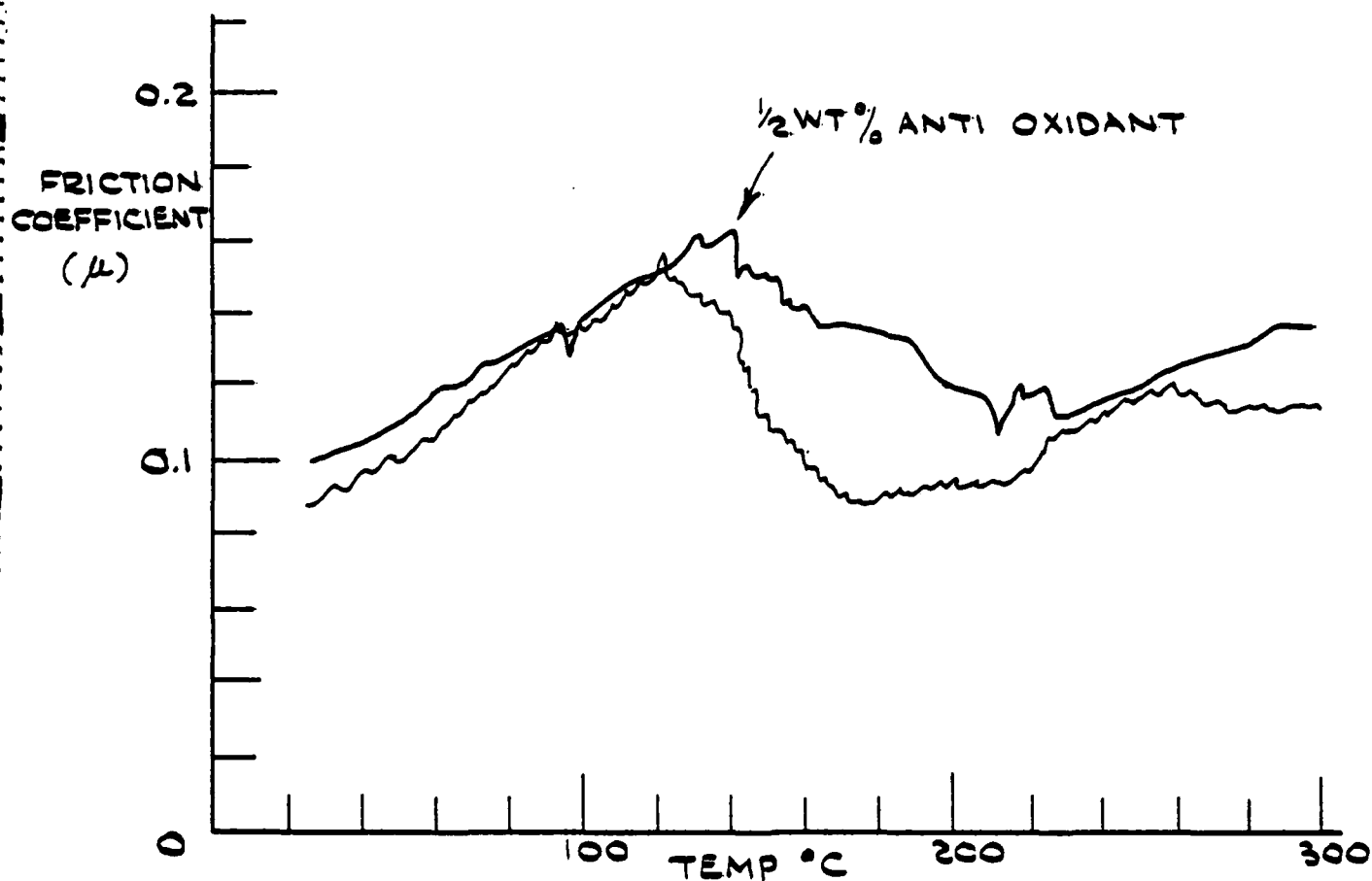


Fig.30. Friction - temperature. Synthetic hydrocarbon with and without anti-oxidant Liner K Immersed.

b) Base Stocks

The foregoing shows the effect that additives in the lubricant have on the AW film. Base stocks, with no extra additive present, do not behave in exactly the same way.

The Volvo base oil, B 1, gave a reasonable ECR in the Anti-wear region, as Figure 26 shows. Note the scale of the resistance graph; here 10 ohms is the mid point. The friction follows the ECR curve closely. A series of 7 hour tests at temperatures from 100°C to 260°C were carried out. Interesting to note is that while the AW region stopped at 170-180°C - indicated both by friction and ECR - the wear stayed low to 220°C. This indicates the existence of an Extreme Pressure region between 180° and 220°C. It was suspected that this was due to the naturally occurring sulphur compounds in the oil.

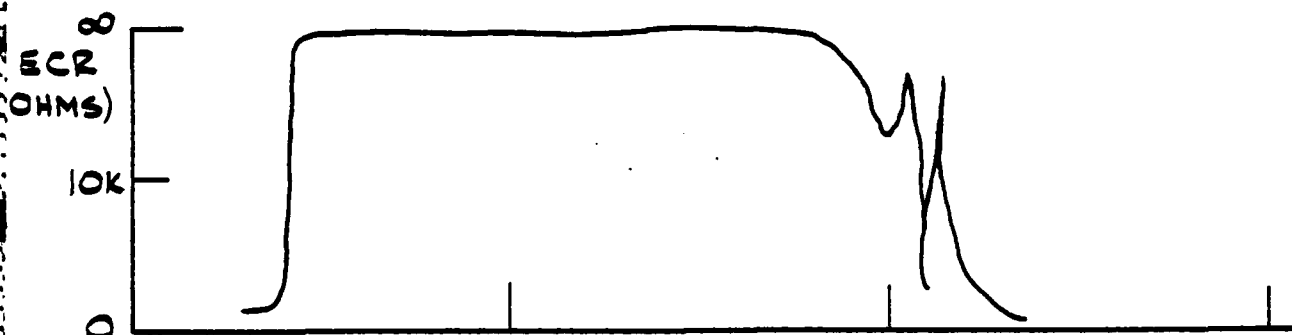
A surprising finding which has not been reported previously is the generation of a small voltage over some region of the temperature range. This is shown on Figure 26. It is also indicated in the subsequent two graphs. At this stage this phenomenon is merely reported and clearly should be the subject of further work.

Next, the 350 NS was studied (Fig. 27). An interesting feature is the ECR trace from room temperature to 80°C. This exhibits the metallic soap formation described by Fein and Kreuz (7) which failed at 80°C as predicted. From there on the friction and ECR behave in step. In the region above 150°C the friction drops, due possibly to formation of active materials by oxidation which progressively fail leading to a gradual friction increase starting at ~ 220°C accompanied by an uncertain ECR trace and a step increase in wear.

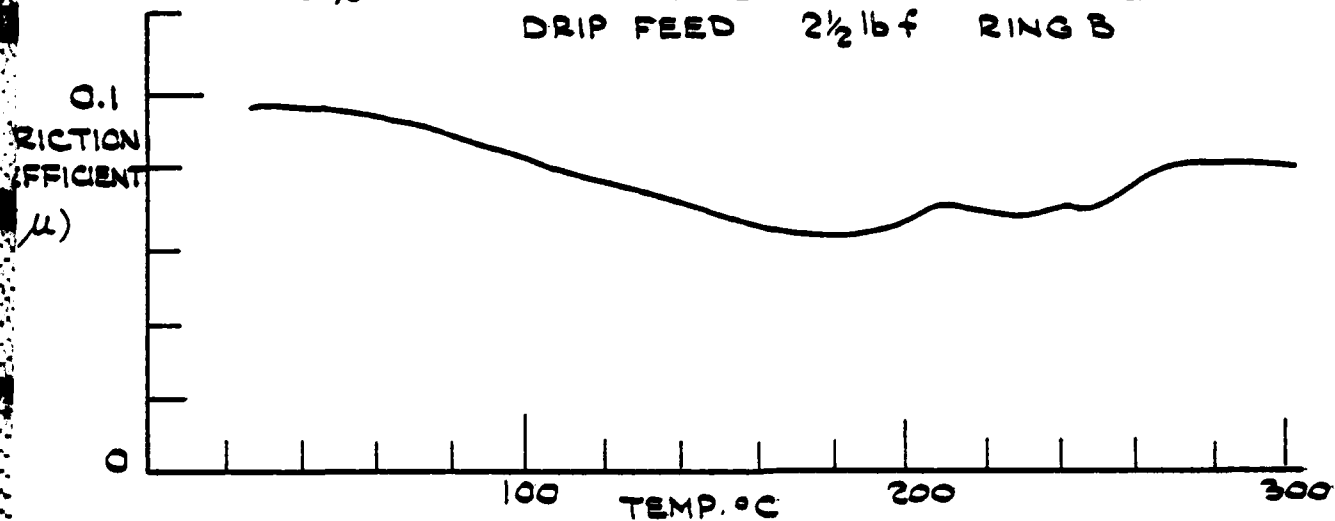
Percolating the NS350 through active alumina ( $Al_2O_3$ ) to remove surfactants reduced the "Fein & Kreuz" region but surprisingly gave an appreciable ECR between 70° and 160°C (Fig 28). It was possible that the naturally occurring sulphur compounds in the oil play a considerable part. A few experiments were carried out with a synthetic lubricant (details in the appendix) which contains no sulphur. Both the friction and ECR traces were quite different as Figure 29 shows.

To study this further, 1/2% of a hindered phenol anti-oxidant, Topanol, a radical trap, was added to this synthetic. The friction in the 120-220°C zone was significantly higher, Figure 30. This whole question clearly needs further research.

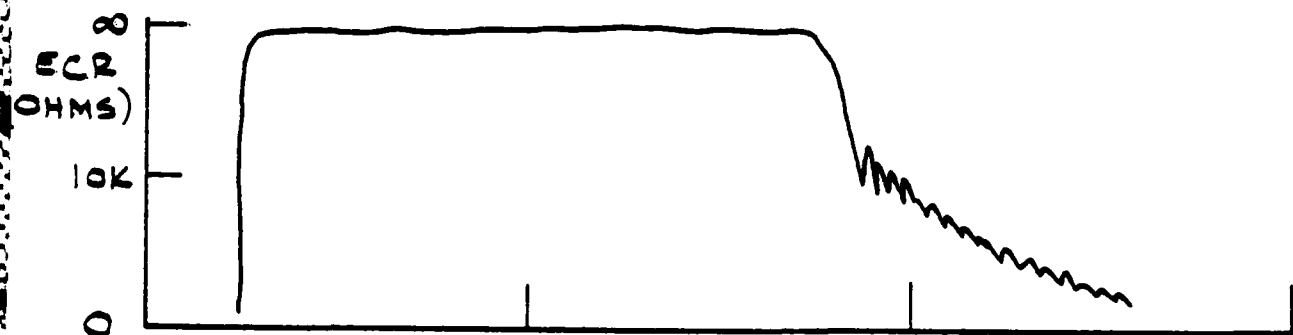
The breakdown of the high ECR anti wear film is for any given system influenced mainly by temperature, but to some extent load has been found to affect it. Its formation was (surprisingly) also a function of the amount of calcium detergent present. This will be discussed next.



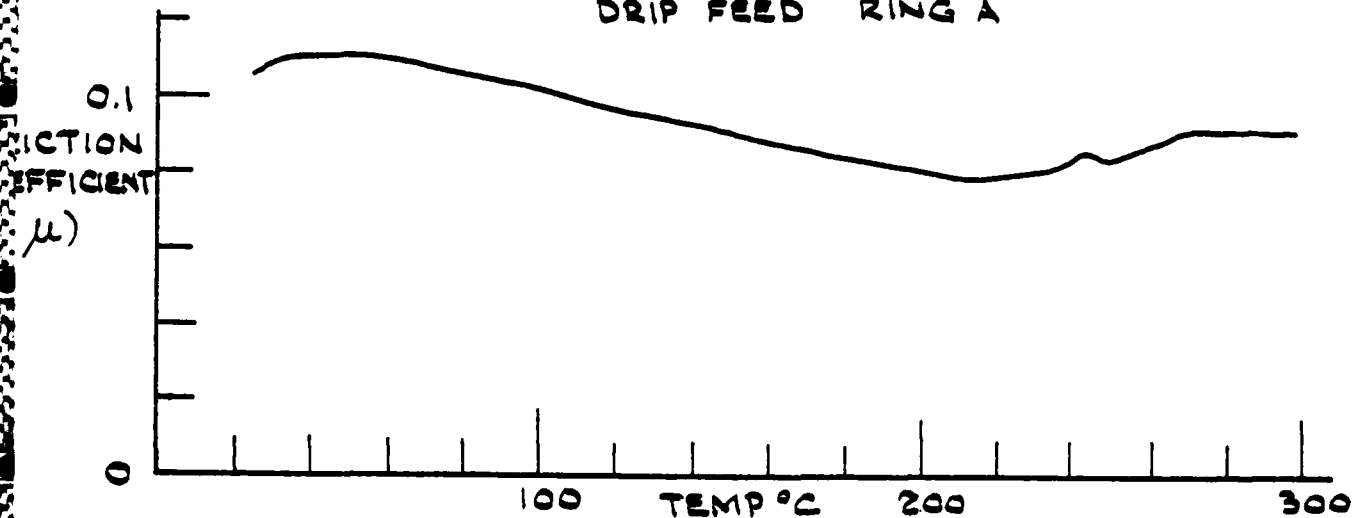
5% CA DETERGENT B1 TORNADO LINER  
DRIP FEED 2½ lbf RING B



31. Friction and ECR - Temperature 2½ lbf 5½ Ca in B1 Tornado liner Ring B



5% CA IN 350NS LINER K 2½ lbf  
DRIP FEED RING A



32. Friction and ECR - Temperature 2½ lbf 5½ Ca in 350 NS Liner K Ring A

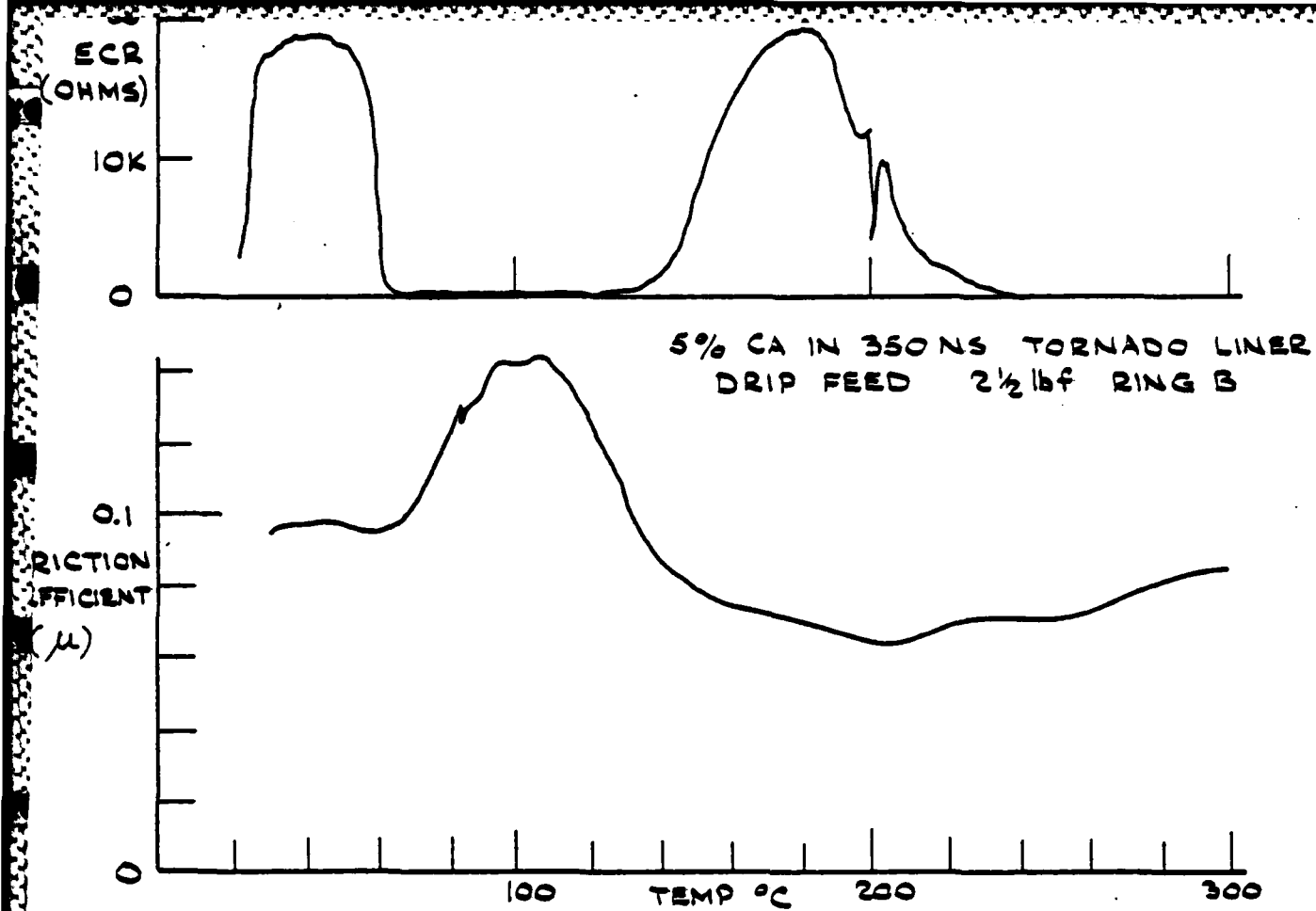
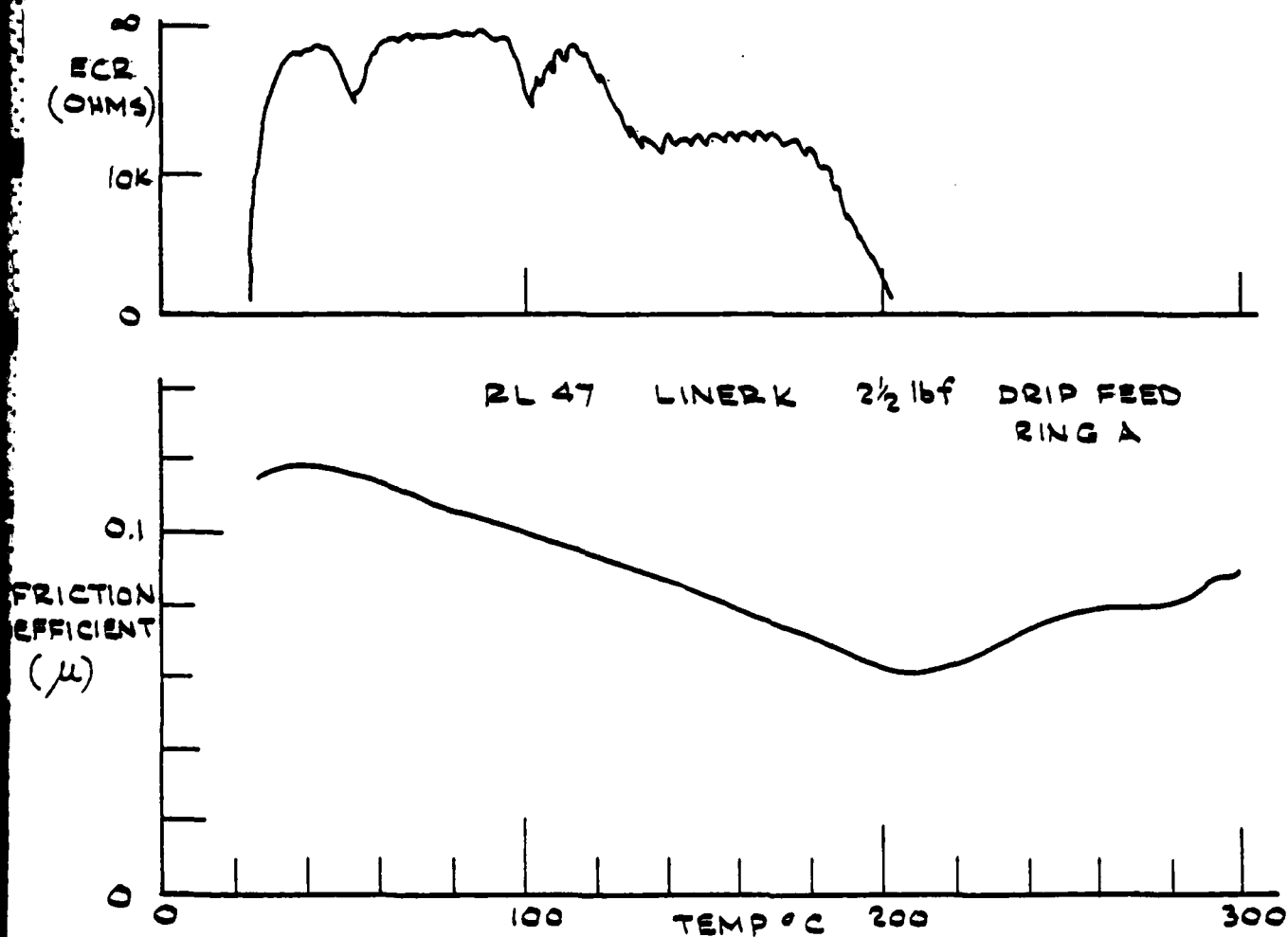


Fig.33. Friction and ECR - Temperature 2½ lbf 5½ Ca in 350 NS Tornado liner Ring A



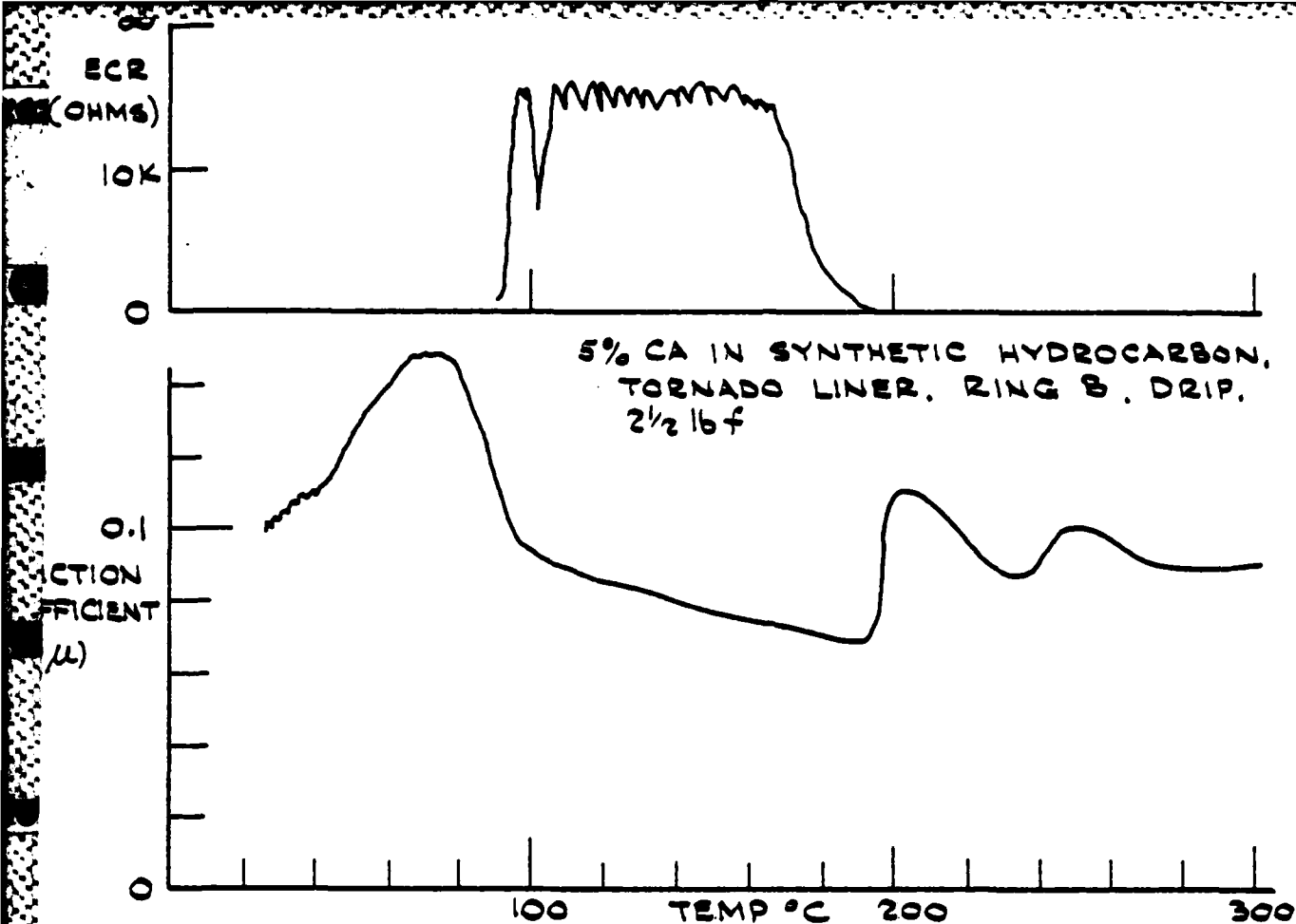
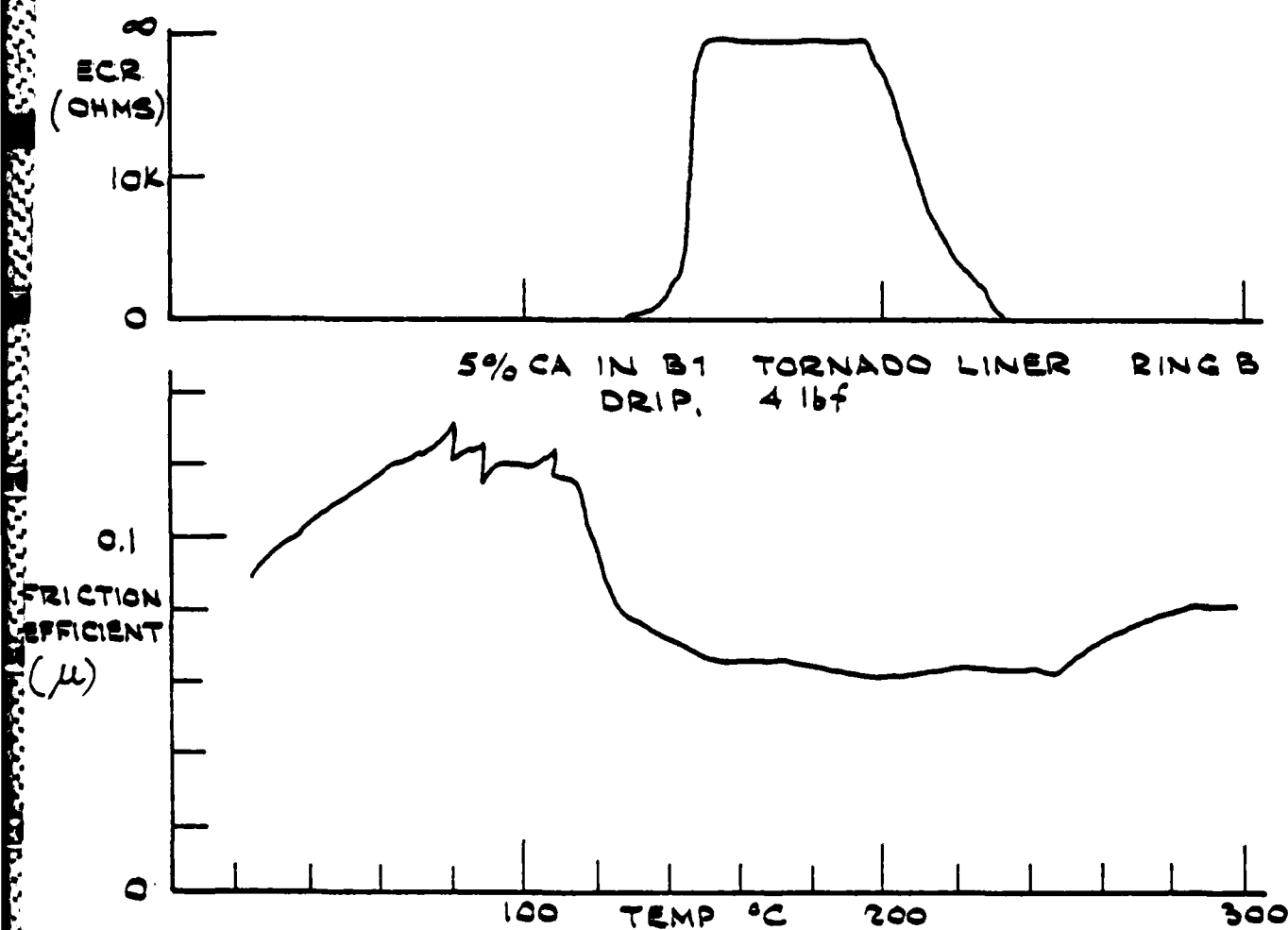


Fig. 35. Friction and EC - Temperature 2 1/2 lbf 5 1/2 Ca in Synthetic hydrocarbon tornado liner Ring B



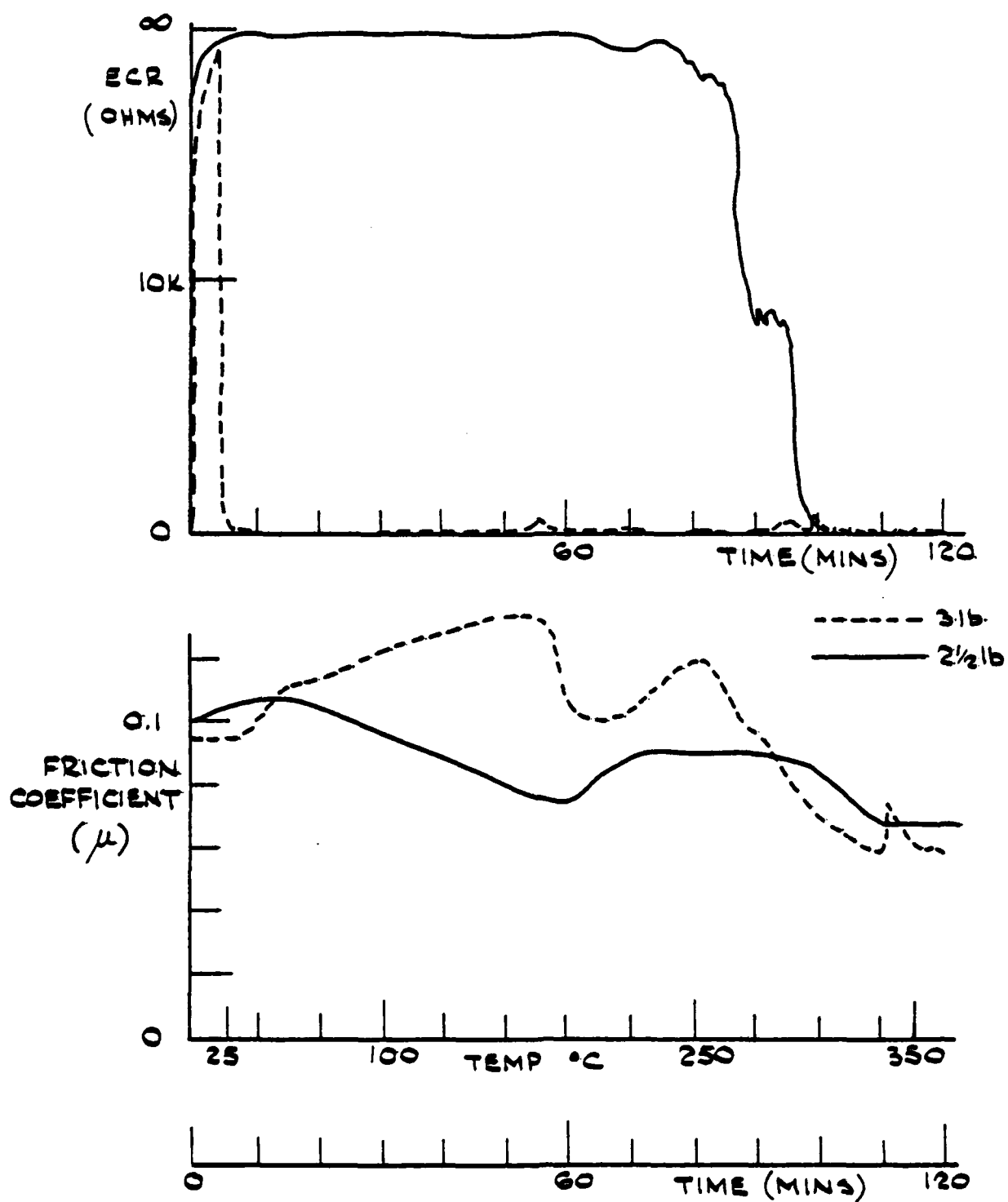


Fig. 37. Friction and ECR - Temperature 2 1/2 lbf 5 1/2 Ca in 350 NS Tornado liner loads 2 1/2 lbf and 3 lbf Ring A



### Effect of Calcium Detergent

It was found that calcium detergent influenced the formation of the AW film and the amount of wear. This was found in another test when the %wt of calcium in the various test oils was plotted against wear (Figs. 15, 22 and 23).

In a further set of tests 5% of a calcium detergent whose properties are given in the appendix, was added to some of the base oils used in the tests. Liners from the Tornado and liner K were used. The base oils were B1, NS350 and the synthetic hydrocarbon. The loads in these figures were 2½lbf. One test, oil B1 with a Tornado liner, was at 4 lb.

In all cases the very high ECR, which was of the order of  $10^6$  ohms, AW region, failed at 180°-200°C. In two cases oil B1/Tornado (Fig. 31) and 350NS/liner K (Fig. 32) the friction and AW region were nearly the same. The 350NS/Tornado combination on the other hand showed almost no AW zone (Fig. 33) indicated by low ECR and high friction. The reference oil RL47 with liner K was not very different, as Fig. 34 shows.

The synthetic hydrocarbon/Tornado shows the AW zone most dramatically, friction and ECR running closely in step (Fig. 35).

Finally one test was run at 4lbf load with the oil B1/Tornado combination. The difference between this (Fig. 36) and the 2½lbf run (fig. 31) is clear. No AW film is built up till 140°C at 4lbf, whereas it was formed at 40°C - 100°C lower - at 2½lbf. The friction follows suit. This effect of load is discussed in more detail later.

All this work shows clearly how important calcium is in the Anti-wear region. Calcium is not normally considered as an Anti-wear additive.

### Effect of Load on Anti-wear High ECR Region

In the last section one result was presented. An increase of load from 2½lbf to 4 lbf influenced the Anti-wear high ECR film.

Two tests were carried out using the 350 Neutral Stock, 350NS, with 5% of the 300TBN Calcium detergent added, as used in the previous section. One run at 2½lbf and the other at 3 lbf. The results are shown in Figure 37. In this case the extra load has inhibited the formation of the Anti-wear high ECR layer almost completely.

The effect of load on the AW high ECR film is not easy to understand. A basic assumption in the design of the HFR is that the stress on all asperities is equal, and is independant of the load. It is therefore at first sight difficult to see why load should play a part. A resolution of the difficulty could be as follows.

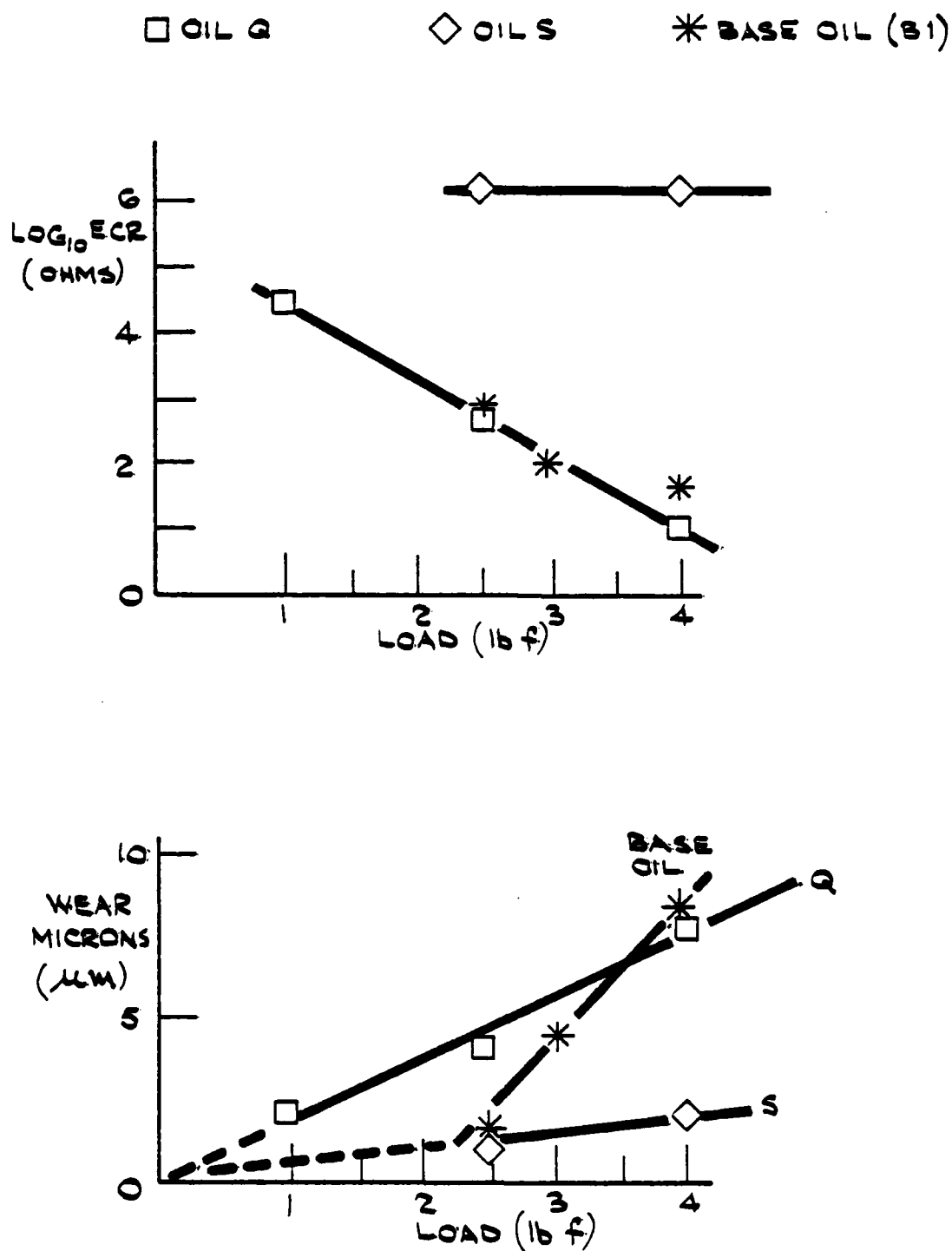


Fig.38. Wear, log ECR - load 100°C Tornado liner Oil Q, S, Base oil

Assume that high ECR indicates that there is a surface film of, say, several microns thickness, and could be an organo-metallic polymeric layer. It will either be solid, or visco-plastic of a high viscosity. There is evidence that in some cases it is solid, as the voltage trace on the cathode-ray oscilloscope stays high at the end of each stroke and indeed stays high for a short time after the machine is switched off. Therefore, if it is a solid film, extra load could bring about its rupture, so leading to asperity contact.

If, on the other hand, the film is viscous or visco-plastic then the oscillation could generate a hydrodynamic film. Some preliminary measurements by Mills (1) suggested that the layer consisted of a very viscous film of several thousand centistokes viscosity. This viscosity would be enough to form a coherent hydrodynamic film. At the end of each stroke (where there is zero velocity) the surfaces are kept apart by a squeeze film. Increasing the load would decrease the film thickness, causing eventual asperity contact. The hydrodynamic film proposed here is generated by the very high viscosity of the visco-plastic layer, not by the bulk viscosity of the lubricant, which at these temperatures is very low indeed.

Finally, some tests were done to discover the effect of load on wear and on the electrical contact resistance. A base oil and the oil with two different additive packages Q and S were studied at 100°C as a function of load. The results are shown in Figure 38.

Oil S gave a very large resistance of  $10^6$  ohms and had a very low wear over a 7 hour test at both  $2\frac{1}{2}$  and 4 lbf. The wear with oil Q increased nearly linearly with load, the logarithm of the ECR fell, also nearly linearly, with load from 1 lbf to 4 lbf. The base oil showed a similar wear to oil S at  $2\frac{1}{2}$  lb, but then the wear rate rose to that of oil Q at 4 lb. The ECR of oil Q and its base oil was nearly the same. These findings can be correlated with the hypothesis outlined above, that load thins the AW film.

These very detailed tests were carried out to obtain an insight into the mechanism of the AW region. They have certainly highlighted the complexity of the processes involved, and give a useful base from which to start further work.

#### Effect of Load on EP

During the scuffing tests for the 4 US Army oils, it was established that the critical scuff temperature of 316°C (Fig. 23) was independent of load at least in the range  $\frac{1}{2}$  lbf to 6 lbf. This differs from the work described in the previous section, where load played a part in causing the AW film to fail.

Seizure between surfaces is prevented if there is a suitable film interposed between the contacting asperities preventing them from seizing. This film is usually chemically formed during the out-of-contact period and substantially destroyed during contact, (8). This means therefore that the formation of the protective layer must be fast enough to allow it to reform before each contact. A mechanism of this type

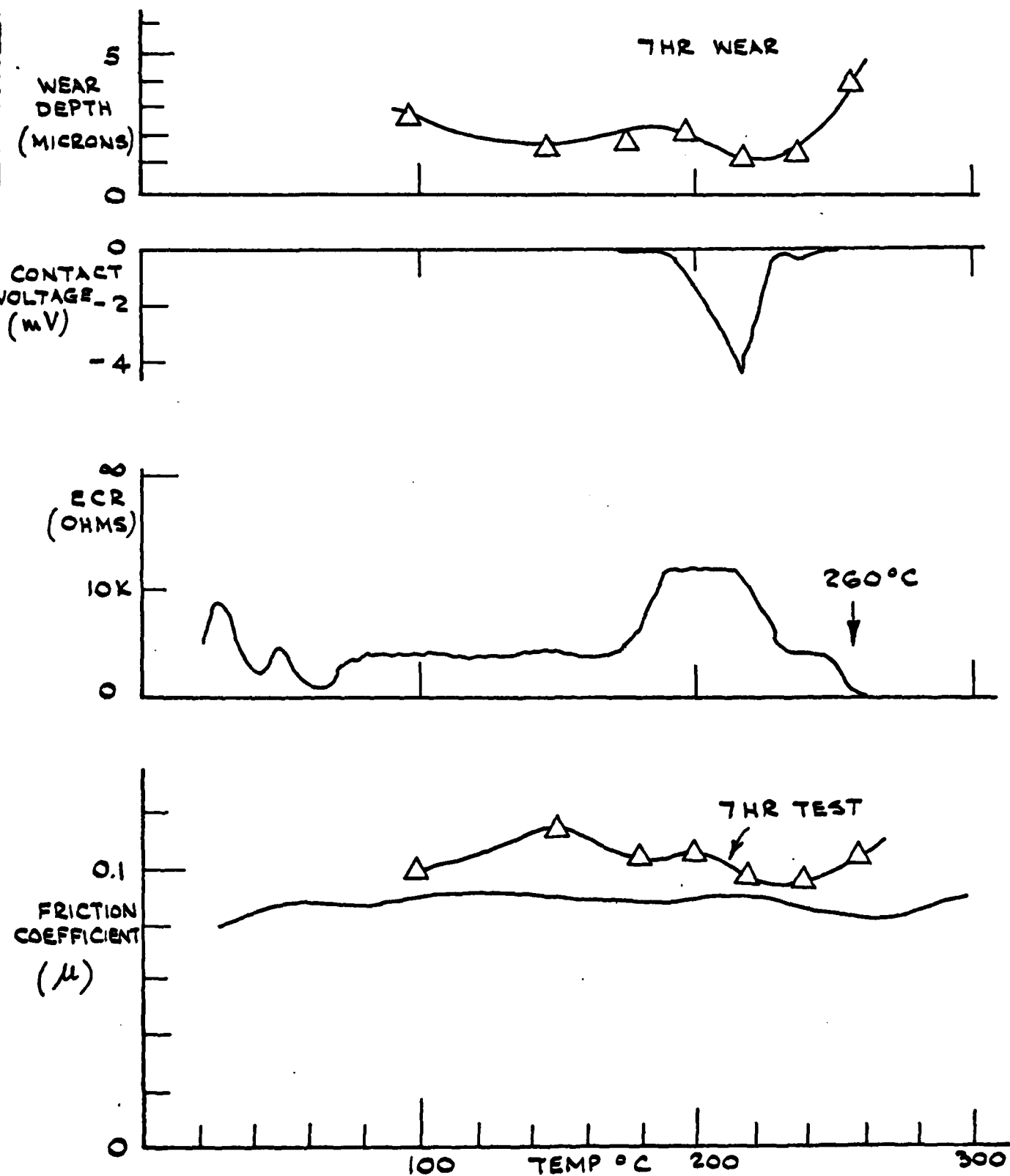


Fig.39. Wear and Friction at 7 constant temperatures Castrol oil T  
Tornado liner

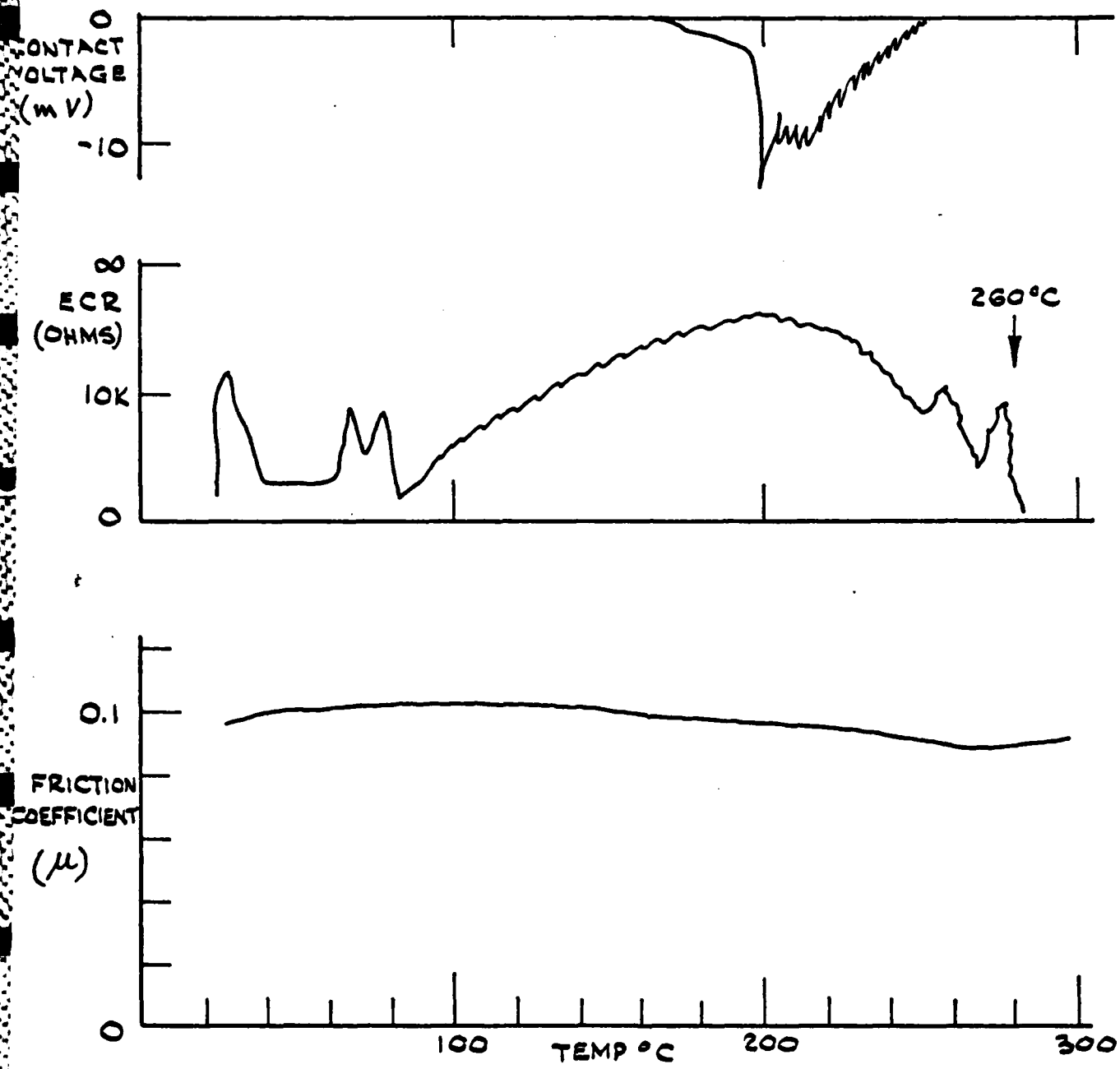


Fig.40. Friction. Contact Voltage - temperature Castrol oil T Volvo liner

# STANDARD LINER

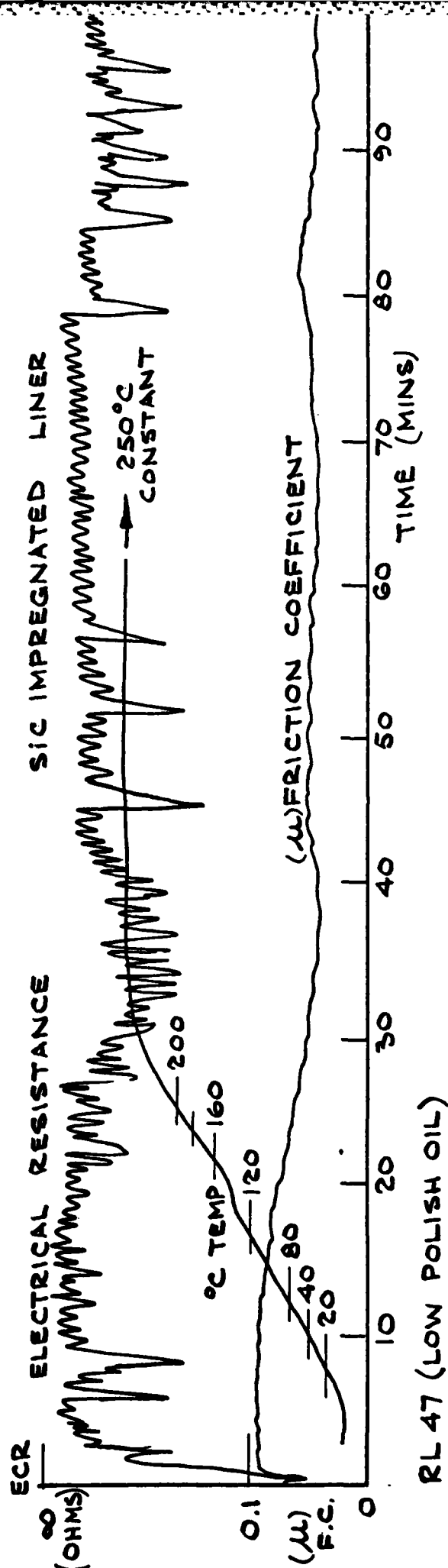
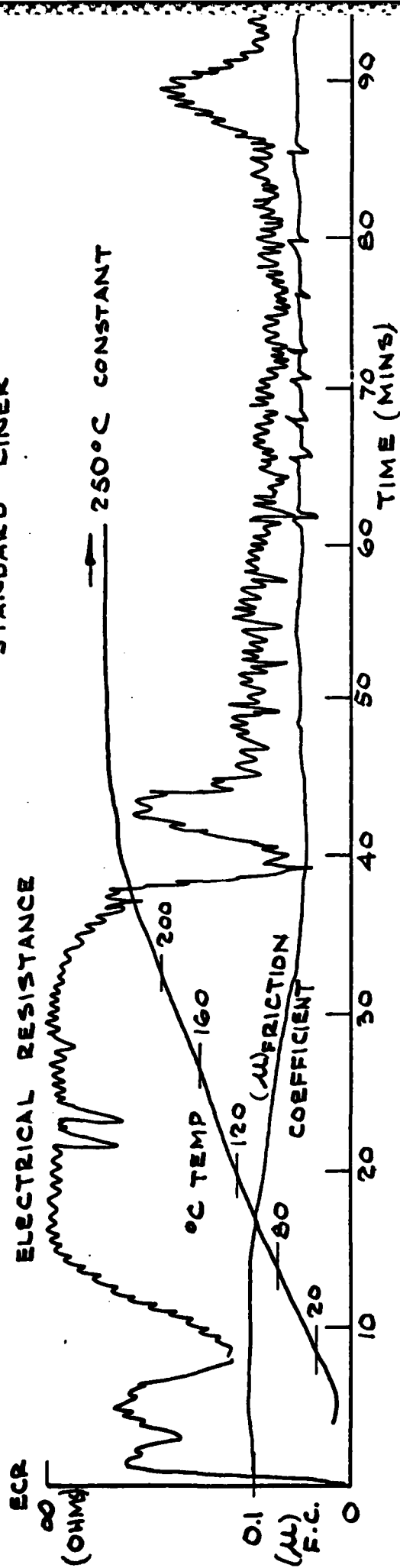
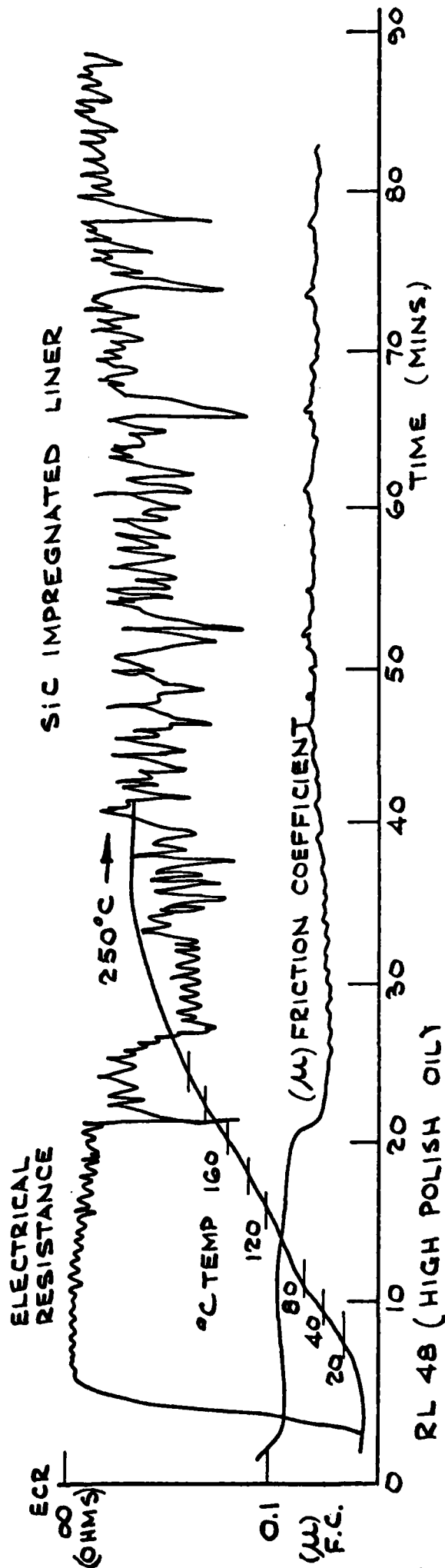
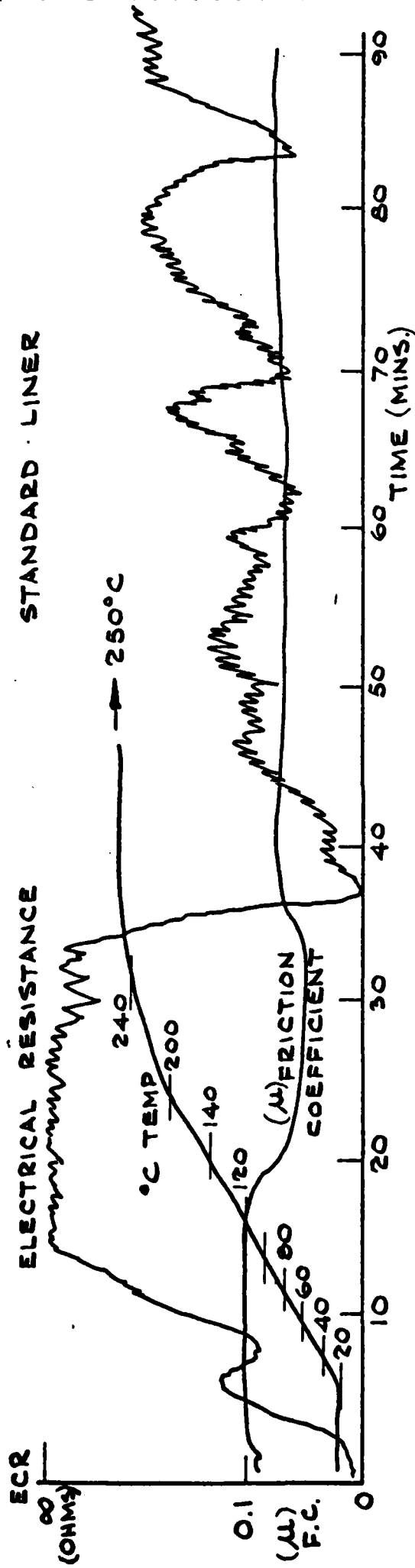


Fig. 41. Recorder traces. Friction, temperature, ECR - time RL 47 oil



RL 48 (HIGH POLISH OIL)

could explain why, even though an AW film is not formed, seizure does not take place. The 3lbf test shown in Figure 37 is an example of this mechanism.

In the case of the 4 US Army oils the rate of formation of the protective layer at this high temperature is fast enough to ensure the surfaces are separated. Load does not enter into this process.

#### PART 4. LINER METALLURGY AND FINISH. COMBUSTION ATMOSPHERE

##### Introduction

The dramatic (and unexpected) effect of liner material was briefly studied using the model oil with the Tornado and Volvo liners. The results are shown in Figures 39 and 40, where the only difference between the two tests is that in Figure 39 a Tornado liner was used, and in Figure 40 a Volvo. Again the self generated contact voltage is shown.

No conclusions can be drawn from these two tests other than to show once more that the HFR machine does discriminate between materials.

##### Silicon Carbide Impregnation

Further tests were carried out on two liners prepared by the Rolls Royce Diesel division. Both were made from the same cast iron. One liner was finished by the standard honing process; the other had undergone a silicon carbide SiC slurry impregnation (Laystall process, Ref. 9). Reference oils RL47 and RL48 were the lubricants tested in a 2 x 2 matrix.

The results are shown in the form of recorder traces (Figs. 41 and 42). What is so interesting is that the use of silicon carbide slurry causes a considerable change in the friction and ECR. Scanning electron microscope pictures and EPMA tests failed to reveal any substantial amount of silicon carbide on the surfaces, yet there was a significant change in performance. In Figure 41 the lubricant was RL47, the low polish oil. The liner treated by the silicon carbide slurry showed a much greater electrical resistance throughout. The friction followed the ECR in both cases. The difference between the two finishes was more marked with the high polish oil, RL48. Both showed a drop in friction in the 120°-160°C region. The slurry finish kept the friction low as the temperature increased to 250°C, but with the standard honed liner, it rose. This rise was associated with a drop in electrical contact resistance, as had been found in other tests.

##### Used Oil and Liner

Finally, four experiments were carried out, again on a



- 1. NEW LINER - NEW OIL
- 2. REHEATED LINER - NEW OIL
- 3. NEW LINER - REHEATED OIL
- 4. REHEATED LINER - REHEATED OIL

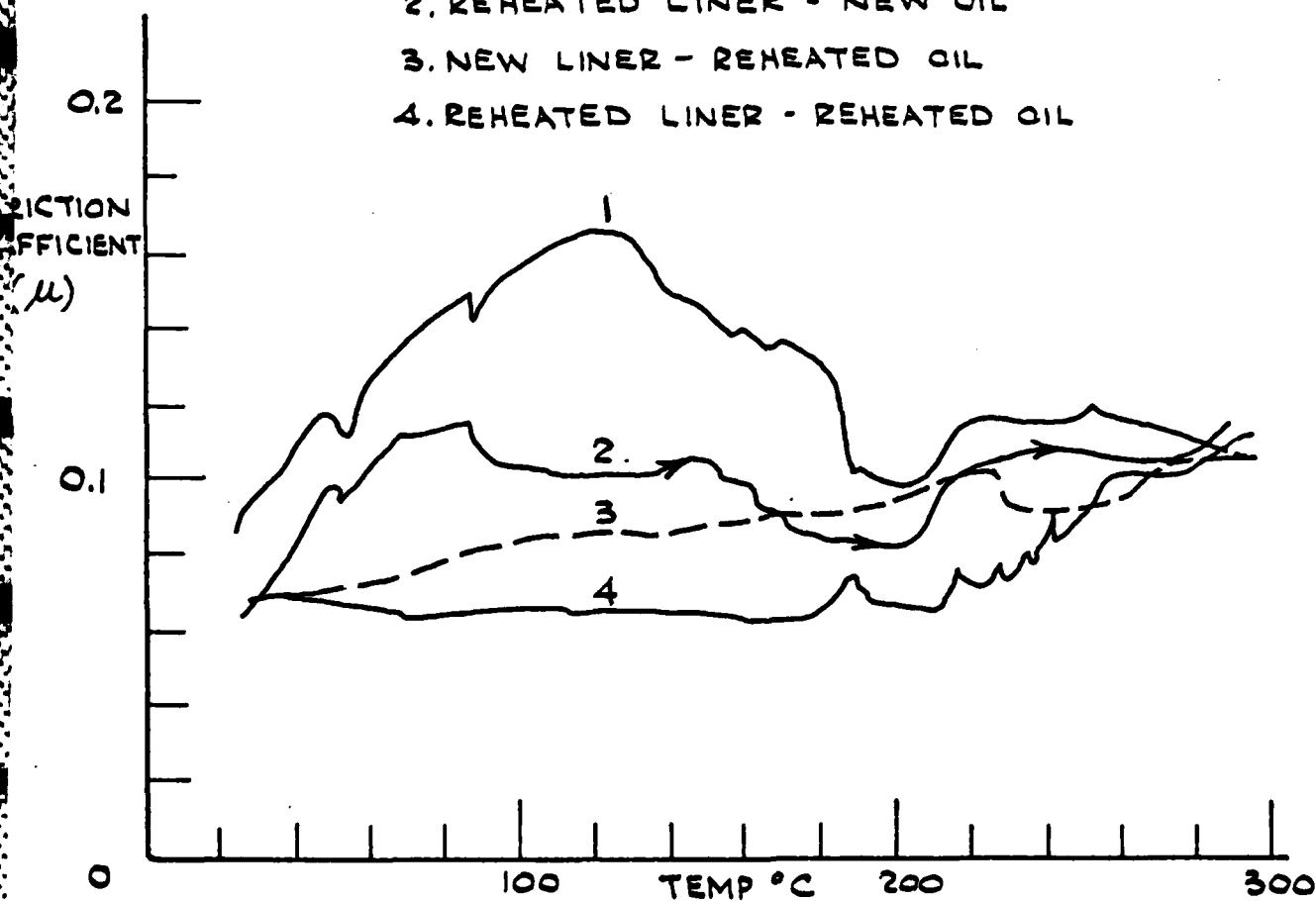


Fig.43. Friction - temperature. (i) New liner, new oil  
(ii) Reheated liner, new oil  
(iii) New liner, reheated oil  
(iv) Reheated liner, reheated oil

2 x 2 matrix, with liner K at  $2\frac{1}{2}$  lbf load using the synthetic oil as lubricant. In this case the lubricant was in the immersed condition. First a new liner segment was run with new oil up to 300°C. In test (ii) the liner was taken out, washed ultrasonically in toluene and acetone as normal, then replaced and run with fresh oil. Test (iii) used a new liner with oil which remained in the bath after test (ii). Test (iv) was simply a re-run of test (i) with used oil and a used liner. All these four tests are shown in Figure 43, labelled (i) to (iv).

This series is reported to indicate the sensitivity of the frictional system to oxidation.

#### Combustion Atmosphere

Finally the question of test atmosphere must be considered. The one major feature of engine lubrication not reproduced in the HFR is the engine cylinder combustion atmosphere. It is not difficult to enclose the specimens in a sealed housing and feed the exhaust from a diesel engine into it. It was realised however that free radicals are a major component of the atmosphere in the cylinder of a running engine. These have almost entirely recombined by the time the gases reach the exhaust manifold and so the exhaust gas is very different from the gas in the combustion chamber. Therefore using it to blanket the test specimens would not be of significant value.

The main requirement is of course to simulate the effect of sulphur in the fuel which has been shown (10,11) to affect bore polishing, sometimes significantly. It was suggested that this could be achieved by neutralising a fraction, say  $\frac{1}{3}$ , of the alkalinity (Total Base Number, TBN) in the oil by sulphuric acid. This was done and a preliminary test was carried out. It was found that the wear increased dramatically. A more detailed study of the effect of sulphuric acid requires further research.

While the question of combustion gases is clearly important, it is seen from all the foregoing work that the HFR machine does correlate bore polishing and scuffing in diesel engines to a very satisfactory degree of accuracy. This would suggest therefore that a combustion atmosphere is not of prime importance and that it is the state of the lubricant itself that is decisive.

It is difficult to see how this problem of combustion atmosphere can be solved. Possibly it is not of great moment, as the correlation seems to be adequate. It is certainly a question for which a solution would be welcome.

### Conclusions

1. The first conclusion and the most important is that the HFR bench test rig has been shown to reproduce the conditions found in full scale engines. This has been shown firstly by the machine correlating the scuffing performance of 4 oils run by the US Army in the 240 hour 6V-53T scuffing test. Secondly, the results of 11 different oils which had been tested in the 200 hour Tornado bore polish sequence were successfully ranked. Of these eleven oils, 5 had also been run in the Volvo TD120A test. These were also rated correctly.

2. The second conclusion is that in the Anti-wear region a layer is formed having a very high electrical contact resistance (ECR) which may reach  $10^6$  ohms. This had not been identified with bore polishing. It is clearly of considerable practical importance and should be investigated further.

3. Finally, the influence of liner material and finish can also be studied on this machine.

### Acknowledgements

The authors would like to thank the US Army Research Office for financial support. The Army Fuels & Lubricants Laboratory, San Antonio, Texas, sent six oils, four of which had been rated in the 6V 53T test, together with liners from the engine. Mobil Oil, Ltd. of London provided six oils which they had tested in the Ford Tornado engine test. Castrol Research Laboratories supplied five oils rated in both Tornado and Volvo tests, together with their base oil. Rolls Royce Diesel division made the tests on two liners possible. Without this help the project could not have been carried out. All these the authors would like to thank.

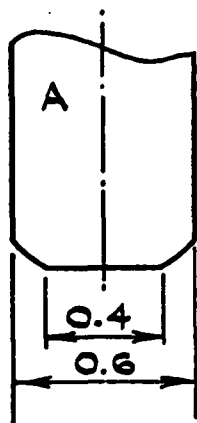
## References

1. T.N. Mills & A. Cameron . Basic Studies on Boundary EP and and Piston Ring Lubrication using a Special Apparatus. ASLE Trans. 1982 25 117-124.
2. S.G. Pumpfrey & A. Cameron. Assessment of Bore Polishing Properties of Lubricating Oils by Bench Tests. Technische Akademie. Esslingen Conference. Testing & Evaluation of Engine Oils. June 1983. pp 7.1-7.10 - also -  
S.G. Pumpfrey & A. Cameron. The use of the High Frequency Rig to Simulate Diesel Bore Polishing. Leeds/Lyon 9th Symposium. Sept.1982. pp 176-179.
3. P.H. Cann, H.A. Spikes & A. Cameron. Thick Film Formation by Zinc Dialkyl Dithio Phosphates. ASLE Trans. 1983 26 48-52.
4. G.M. Hamilton & S.L. Moore. The Lubrication of Piston Rings. Proc. Inst. Mech. Eng. 1974 188 pp 253-268
5. J.A. McGeehan. A literature review of the Effects of Piston and Ring Friction & Lubrication, Oil Viscosity & Fuel Economy. SAE Tech. paper 780673. 1978.
6. W.G. Fiennes & J.C. Anderson. Electrical Conduction in Thin Lubricating Oil Films. Radio and Electronic Engineer 1974 44 141-152
7. R.S. Fein, K.L. Kreuz & S.J. Rand. Solubilization Effects in Boundary Lubrication. Wear. 1973 23 393-407
8. M.W. Bailey & A. Cameron. The Influence of Temperature and Metal Pairs on the Scuffing of a Commercial Oil. Proc.Inst.Mech.Eng. 1973 187 757-761
9. John E. Tanner. Silicon Carbide Reduces Bore Wear. SAE Tech Paper 830069 1983
10. J.A. McGeehan. Effect of Piston Deposits, Fuel Sulfur and Lubricant Viscosity on Diesel Engine Oil Consumption and Cylinder Bore Polishing. SAE Tech Paper 831721 1983
11. J. Ayel, G. Tahon, J.C. Fayard. Mecanisme de l'usure par Polissage des Cylindres de Moteurs Diesel. Institut Français de Petrole. Publication no. 29996 March 1982.

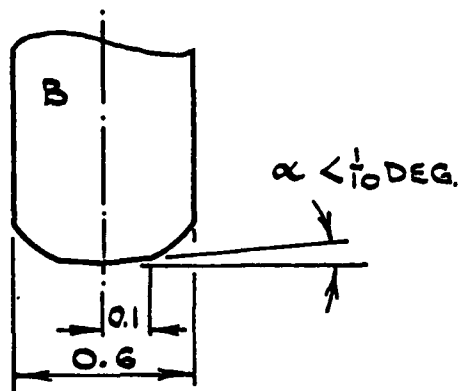
List of Figures

1. Schematic Drawing of Test Rig.
2. Instantaneous Trace of Friction.
3. Circuit to measure Electrical Contact Resistance. ECR.
4. Modified Test Rig.
5. Liner and Wear Scar.
6. Surface Roughness and Wear Scar Depth.
7. Friction and Electrical Contact Resistance, ECR. RL47 and RL48,  
160°C Liner K 2lbf Load
  - a) 2 hour. Immersed
  - b) 3 hour. Drip Feed 1.6 ml/hour.
8. Wear Scar from previous tests (Fig 7)
9. Wear - time. RL47 and RL48. Wear 2 Liner K load 2½lbf 180°C  
Drip Feed.
10. Load - wear. RL47 and RL48. 7 hour test Wear 2 Tornado liner.  
180°C. Drip Feed.
11. Friction - time. Loads ½, 2, 2½lbf. Heating rate 7°C/min to  
180°C. Tornado liner RL47 Wear 2 Test duration 7 hours. Ring A.
12. Friction - time. Loads ½, 2, 2½lbf Tornado liner RL48 Drip feed  
Wear 2 Test duration 7 hours.
13. Wear - temperature. Load 2½lb 7 hour test. Tornado liner  
RL47 and RL48 Drip feed Wear 2.
14. Friction and ECR - Time. Load 2½lb 7 hour test Drip feed 3.3 ml/hour  
Wear 2 RL47 and RL48 Tornado liner. Ring A.
15. Calcium in oil wt% - Wear. US Army oils and RL47 and RL48  
US Army liner Wear 2 7 hour test. Ring A.
16. % Bore Polish - wear. Tornado liner. 6 Mobil oils. Wear sequence 2
17. % Bore Polish - wear. Tornado liner. 6 Mobil oils. Wear Sequence 3
18. Wear - Temperature. Oils Q, S, T. Base Oil Tornado liner. Ring B.
19. Log Electrical Resistance - Wear depth from Fig. 18.
20. Wear - Tornado Bore Polish. Tornado liner
21. Wear - Volvo Bore Polish. Volvo liner.
22. Calcium in Oil - Bore Polish
  - a) Volvo
  - b) Tornado
23. Calcium in Oil - Wear depth. Cross plot of Figs. 21 and 22a)

24. Friction and ECR - Temperature US Army liner Oil AL 6855L Ring A
25. US Army Scuffing Test 6V 53T - Critical Temperature Oils A B E F
26. Friction, ECR, Contact Voltage, Wear - Temperature Oil B1  
Tornado liner
27. Friction ECR Contact Voltage - Temperature 350 NS Tornado liner Ring B
28. - ditto - Purified 350 NS Tornado liner
29. - ditto - Synthetic hydrocarbon  
Tornado liner
30. Friction - temperature. Synthetic hydrocarbon with and without  
anti-oxidant Liner K Immersed.
31. Friction and ECR - Temperature 2½lbf 5% Ca in B1 Tornado liner Ring B
32. " " " " " " 350 NS Liner K Ring A
33. " " " " " " 350 NS Tornado liner Ring A
34. " " " " " " RL47 Liner K Ring A
35. " " " " " " Synthetic hydrocarbon  
Tornado liner Ring B
36. " " " " " " Oil B1 Tornado liner
37. " " " " " " 350 NS Tornado liner  
Ring A loads 2½lbf and 3lbf
38. Wear, log ECR - load 100°C Tornado liner Oil Q,S, Base Oil
39. Wear and Friction at 7 constant temperatures Castrol oil T Tornado liner
40. Friction. Contact Voltage - temperature Castrol oil T Volvo liner
41. Recorder traces. Friction, temperature, ECR - time RL47 oil  
Top) Standard liner Bottom) Silicon Carbide treated liner
42. Recorder traces. Friction, temperature, ECR - time RL48 oil  
Top) Standard liner Bottom) Silicon carbide treated liner
43. Friction - Temperature (i) New liner, new oil, (ii) Reheated liner, new oil,  
(iii) New liner, reheated oil, (iv) Reheated liner, reheated oil.



CONTACT FACE 0.4mm



CONTACT FACE 0.1mm

DETAILS OF RINGS

## APPENDIX

-

## DETAILS OF OILS

Castrol Oils - 15W/40. Approved Daimler Benz, (Sheet 228.1.) based on solvent refined base oil plus additive package to give SHPD performance. The base oil, B1, is a blend of 150 and 500 solvent neutrals with a viscosity at 40° of 66.85 cSt. Density at 15°C 0.890.

	P	Q	R	S	T
<u>VOLVO TD 120 A</u>					
% Polish (RL 47 = 2    RL48 = 14)	12	15	5	3	13

TORNADO

% Polish	15.3	17	13.7	15	11.5
RL 47	22	19	16.5	15	15.4
RL 48	42	39	52	35	57
Corrected % Polish	10	11	8	13	7

ANALYSIS

Sulphated Ash	1.9	1.7	1.9	1.9	1.4
Total Base Number	14	17	15	13	10
Calcium	0.49	0.08	0.51	0.52	0.28
Magnesium	-	0.29	-	-	0.05
Zinc	0.13	0.13	0.11	0.12	0.14

Mobil Oils

Oil	SAE Grade	Sulphate ash content	Major elements	API desig- nation	Tornado Bore polish %
C1 = RL47	20W-40	1.8	Zn,P,Ca	SD/CD	17
C2 = RL48	20	0.95	Zn,P,Ca	SD/CD	45
C3	30	1.95	Zn,P,Ca	SD/CD	15
C4	30	1.0	Zn,P,Ca,Mg	SE/CD	55
C5	15W-40	1.4	Zn,P,Ca,Mg	SE/CD	20
C6	30	1.4	Zn,P,Ca	SD/CD	15

Mobil Synthetic hydrocarbon viscosity at 40°C 29.74 cSt. Density at 15°C 0.817.

Elf-Aquitaine solvent neutral, 350 NS. 103 VI. Viscosity at 100°F 82.34 cSt and at 210°F 9.50 cSt. Density at 15°C 0.884. This base oil was used in the work described in Ref. (1).



US Army Oils

Property	ASTM Method	Lubricant			
		A	B	E	F
SAE Viscosity Grade		30	10W	10W-30	15W-40
Apparent Viscosity at -29°C, cP	D 2602	Solid	12250	3200	23000
Apparent Viscosity at -18°C, cP	D 2602	1000	2180	1600	4200
Kinematic Viscosity at 40°C, cSt	D 445	107	47	56	113
Kinematic Viscosity at 100°C, cSt	D 445	11.6	6.5	10.0	14.8
Viscosity Index	D 2270	96	103	167	135
Total Acid Number, mg KOH/g	D 664	2.3	2.5	3.0	1.9
Total Base Number, mg KOH/g	D 2896	13.9	12.7	6.3	8.2
Pentane B Insolubles, wt%	D 893	0.03	0.02	0.04	0.02
Toluene B Insolubles, wt%	D 893	ND	ND	0.02	ND
Benzene B Insolubles, wt%	D 893	0.01	0.01	ND	0.01
Flash Point, °C	D 92	223	214	224	210
Pour Point, °C	D 97	-17	-31	-45	-30
Gravity, °API	D 287	25.5	28.3	31.2	28.1
Carbon Residue, wt%	D 524	2.1	1.7	1.1	1.0
Sulfated Ash, wt%	D 874	1.6	1.7	1.1	1.0
<u>Elements, wt%</u>		<u>Method</u>			
Mg	AA	0.003	0.002	0.000	0.13
P	XRF	0.11	0.11	0.13	0.12
S	XRF	0.48	0.43	0.33	0.65
Ca	AA	0.46	0.46	0.25	0.10
Zn	AA	0.08	0.09	0.13	0.12
Ba	AA	<0.005	<0.005	<0.005	<0.005
N	CLM	0.07	0.07	0.14	0.12
Na	AA	0.001	0.002	0.001	0.002
Cl	AA	<0.01	<0.01	<0.02	<0.02
AL Code		8980	6855	8924	7172
		6856	OE/HDO-10	8406	
		OE/HDO-30			

ND - Not Determined  
 CLM - Chemiluminescent Method  
 AA - Atomic Absorption  
 XRF - X-Ray Fluorescence

Detergent-Dispersant. Physical and Chemical Properties.Ca Sulphonate  
(TBN 300)

Density at 15°C	1.16
Viscosity at 100°C, cSt	95
Colour dil.	6.5
Flash Point, °C	230
TBN, mg KOH/g	322
Sulphated Ash, wt%	
(Elemental Analysis)	34
Calcium, wt%	12.5
Sulphur, wt%	2.4
Nitrogen, wt%	-

**FILMED**

9-84